PERFORMANCE STUDIES
ON
SPRAY-TYPE FLAT PLATE SOLAR COLLECTORS

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NOMENCLATURE

A	Constant in transmissivity - absorptivity product expression.
A_c	Exposed area of the collector plate.
A_j	Surface area of jth section.
A_s	Specific area of the collector plate.
B	Constant in transmissivity - absorptivity product expression.
b	Tilt angle of the collector with horizontal.
b_{opt}	Optimum tilt angle of the collector.
C	Constant of integration.
C_p	Specific heat of transfer liquid.
C_w	Constant dependent upon tilt
D_i	Inner diameter of the insulated cylindrical tank.
D_o	Outer diameter of the insulated cylindrical tank.
d	Declination of the sun.
f	Effective thermal resistance of outer glass plate relative to other glass plates.
H_T	Total amount of radiation absorbed by inclined collector plate per unit area.
h	Hour angle from the noon.
h_o	Convective heat transfer co-efficient between tank insulation and ambient air.
h_w	Convection co-efficient due to wind on the glass cover.
I_D	Intensity of direct radiation on horizontal surface.
I_{DT}	Intensity of direct radiation on tilted surface.

I_d	Intensity of diffuse radiation on horizontal surface.
I_{dT}	Intensity of diffuse radiation on tilted surface.
I_H	Intensity of total radiation comprising of direct and diffuse radiation on horizontal surface.
I_N	Intensity of normal radiation over a surface normal to the sun rays.
I_P	Intensity of radiation absorbed by the inclined plate.
I_T	Intensity of total radiation on tilted surface.
k	Conductivity of the insulating material.
L	Latitude of the place.
M	Mass of liquid as a system load.
m	Number of months according to the design requirements.
$\dot{m}$	Mass flow rate of circulation of transfer liquid.
n	Number of hours in consideration
n_g	Number of glass covers used.
Q_A	Radiation energy available on horizontal surface.
Q_a	Radiation energy absorbed by the section.
Q_c	Net heat convected out from the section.
Q_l	Total heat loss from the section.
Q_R	Heat energy required to meet the demand.
Q_s	Heat energy stored within the section.
Q_U	Amount of useful energy.
R	Overall heat transfer resistance for storage tank.
R_D	Orientation factor for direct radiation.
R_d	Orientation factor for diffuse radiation.

R_g Heat transfer resistance through the insulating material.

R_f Heat transfer resistance through the air film on the outside of the insulation.

R_i Inner radius of the equivalent spherical container.

R_o Outer radius of the equivalent spherical container.

R_R Orientation factor for reflected radiation.

r Radius of the equivalent spherical container.

S_i Inner surfade area equal for both, cylinder and equivalent sphere.

S_o Outer surface area equal for both, cylinder and equivalent sphere.

T_1 Temperature of liquid at the inlet of the collector.

T_2 Temperature of liquid at the outlet of the collector.

T Resultant temperature of liquid in the storage tank at any moment after onset of cooling period.

T_a Ambient air temperature.

T_i Initial temperature of liquid in the tank before operation.

T_f Final temperature of liquid in the tank at the end of operation.

T_m Mean tank liquid temperature at any instant during operation.

T_j* Mean section temperature after a small finite increment of time $\Delta\theta$.

T_p Mean temperature of the absorber plate.

$T_{\theta=0}$ Temperature of liquid to be stored, at the beginning of the cooling period.

t Insulation thickness of the cylinder.

U_j Overall heat transfer co-efficient of jth section.

V	Volume of liquid to be used as a heat storage medium.
W_j	Thermal capacity of the section.
W_t	Thermal capacity of the tank with liquid.
w	Specific weight of the heat storage medium.
dq	Differential heat quantity.
dT	Differential temperature.
ΔT	Required temperature rise.
$d\theta$	Differential time.
θ	Cooling period for storage tank.
$\Delta\theta$	Finite increment of time.
θ_H	Angle of incidence of direct component on horizontal surface.
θ_T	Angle of incidence of direct component on tilted surface.
ρ	Albedo of the ground surface.
α_j^*	Heat absorbing co-efficient for jth section.
σ	Stefan Boltzman constant.
e_p, e_g	Emissivity and absorptivity of collector plate and glass respectively
η_i	Instanteneous efficiency of the collector.
η_b	Bulk efficiency of the collector

$(\tau\alpha)_D$, $(\tau\alpha)_d$, $(\tau\alpha)_R$ Transmissivity-absorptvity product for direct, diffuse and reflected components of radiation respectively.

Subscripts

i	Indicates value at corresponding month.
j	Indicates value at corresponding hour.

ABSTRACT

A new design of flat plate solar collector working on forced circulation principle, suitable for large scale applications, has been developed and experimented.

Design introduces spray nozzles inside the collector which sprays fluid in the form of fine particles upon the bottom surface of the black absorber plate. The fluid, thus, comes in direct contact with the hot plate, forming a thin layer, which moves downward adjacent to the inclined absorber plate by means of gravity. This enables the heat exchange to take place in an efficient manner.

Two collectors with different nozzle configurations have been fabricated and studied for a water heating system, with 50 litres and 100 litres of water as system load. Effects of various flow rates have also been observed.

Design analysis to determine optimum tilt angle of the collector, area of the absorber plate, size of the hot water storage tank and its insulation thickness, has been presented. Cost estimate of the unit is given and compared with existing collector designs.

It is concluded that "spray-type" flat plate collectors are quite efficient and comparatively, in-expensive and should, therefore, be preferred for commercialization.

CHAPTER 1

INTRODUCTION

1.1 SOLAR ENERGY - A PROSPECTIVE ALTERNATIVE ENERGY SOURCE

Energy is the most significant requirement to meet the needs of any country. With the population and industrial growth world over, the demand for energy has also grown tremendously. Although, it is only about a few hundred~~s~~ kwh per head per year in India, it is about 18 thousand kwh per person per year in the U.S.A. and many parts of the Western Europe. The energy sources presently being used any where, are the conventional ones i.e. wood, coal, oil, gas ~~and~~ hydrogeneration etc. At the present rate of their consumption, it is definite that these fossil~~e~~ fuels will exhaust one day in not too distant a future and the mankind may face a serious energy problem. Also, the pollution caused by the use of fossil~~e~~ fuels has interfered with the enveloping atmosphere bringing about changes in the pattern of climatic cycles. This has caused serious health problems in most of the industrialised parts of the world. It is, thus, necessary to look for alternative energy sources and to exploit them for various applications. Solar energy, geothermal energy, wind energy, energy from tidal waves are the major alternative energy sources freely available, normally inexhaustible and relatively pollution free.

We shall concentrate here on the use of solar energy, the available amount of which on the earth, theoretically, far exceeds the world future energy needs.

1.2 SOLAR RADIATION

Quantitative knowledge of solar radiation reaching the surface of the earth is prerequisite for meteorological and climatological studies, and for proper utilization of solar energy. Solar radiation comprises of very wide electromagnetic spectrum, emitted by the sun, at the outer surface of the atmosphere, consisting of ultraviolet radiation having wavelength range of about 0.2 to 0.4 microns, visible radiation between 0.4 and 0.7 microns and infrared radiation with higher wavelengths, (Fig. 1.1). Of the total solar radiation, 9 percent occurs in the ultraviolet, 45 percent in the visible and 46 percent in the infrared region. Maximum intensity occurs in the visible range, (1).

The intensity of solar radiation normal to the sun's rays at the outer limit of the atmosphere varies with the earth-sun distance. At the mean distance, its value has been determined to be 2 cal/cm^2/min with a probable error of 2.0 percent, which is known as the solar constant, (2).

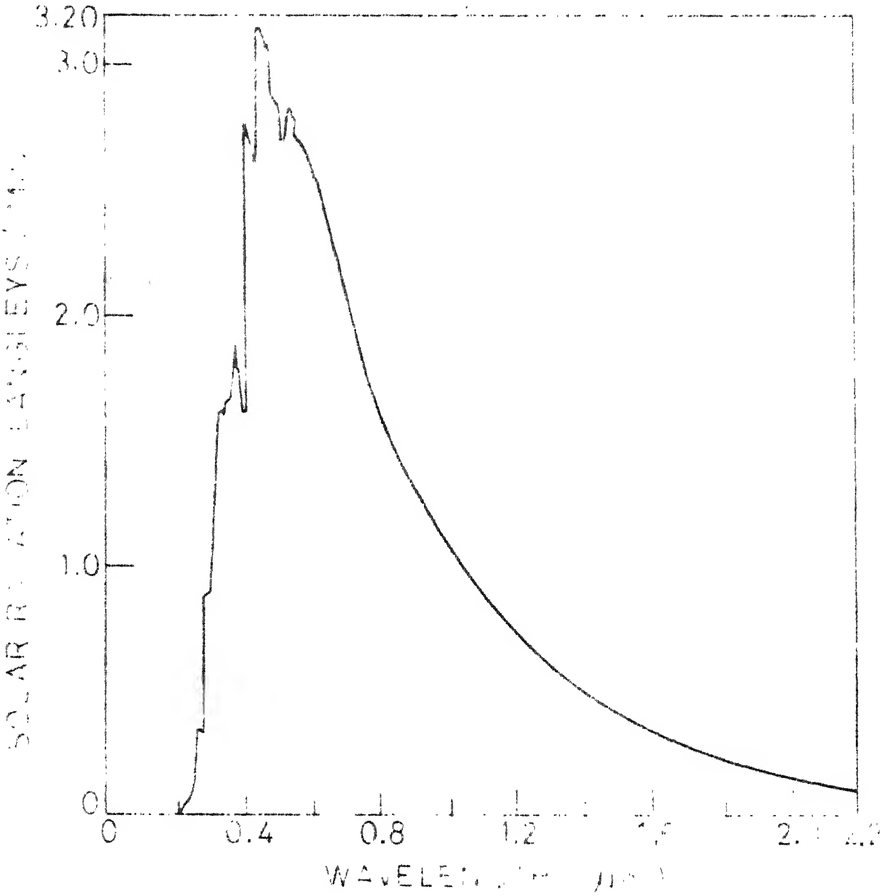


FIG. 1.1 SPECTRAL DISTRIBUTION OF SOLAR RADIATION INCIDENT UPON A SURFACE NORMAL TO THE SUN RAYS AT THE OUTER LIMIT OF ATMOSPHERE.

The factors that deplete the solar radiation in its passage through the earth's atmosphere are (i) scattering due to aerosols, dust and other particles and (ii) absorption by ozone, in the upper atmosphere and by water vapour near the earth's surface. The remaining portion of the original direct radiation may reach the earth unchanged in wavelength.

The scattered solar energy gives rise to the diffuse radiation, which is the radiation coming from the entire sky vault. Whereas, the direct solar radiation has spectral characteristics determined by the solar spectrum and absorption of the atmosphere, the diffuse solar radiation has a spectral distribution, determined by the scattering characteristics of the atmosphere, normally of shorter wavelengths. Thus, a surface on the earth receives solar radiation of two forms - direct radiation and diffuse radiation.

1.3 HARNESSING OF SOLAR ENERGY

There are three technological processes by which solar energy can be utilized,(3), (a) heliochemical, (b) helioelectrical and (c) heliothermal. The first process through photosynthesis maintains life on this planet, the second process using photovoltaic converters, supplies power for all of the communication satellites and the third process provides much of the thermal energy, in the

form of high and low grade energy. Research and experimental work over past few years have found numerous applications of solar energy which can satisfy a great variety of our needs.

Generally, two major problems arise when we attempt to use this energy. The first is caused by the low density (or low flux) of solar radiation, and the second is due to its intermittent nature. Most solar energy applications, therefore, require some form of energy storage for periods of no-solar radiation. In fact, the cost of using solar energy is largely the cost of overcoming these two problems.

Most of the systems, which utilize solar energy, first collect it as heat by means of a collector, which forms a unique and essential component of all systems. This intercepts solar radiation, converting the radiation to thermal energy, and transfers this heat to a working medium. Collection process is based on either of the two basic concepts, i.e. focussing collector or flat plate collector.

1.4 COLLECTORS

Focussing collectors are used to obtain higher temperatures in various applications. This collector employs a reflecting surface having a shape of particular

configuration such as paraboloid of revolution, parabolic cylinder, hemisphere, circular cylinder, plane and conical mirrors etc. The selection of these depends on specific application. After receiving the direct component of solar radiation, the reflecting surface of a collector concentrates the incoming radiation at its focal point or focal plane yielding a high intensity of radiation capable to produce high temperatures ranging between 200 °C and 3500 °C.

An important consideration with focussing collectors is the need for continuous sun-tracking devices in order to get parallel and direct sun-rays, thereby, requiring very complicated and costly arrangements. A significant fraction of the total energy from the sun is diffuse or sky radiation, which is non-focussable and, therefore, lost to concentrators, but is available to flat plate systems. Another limitation which restricts the use of focussing collectors is the necessity of high quality reflecting surfaces.

Flat plate collector, as its name implies, generally, consists of four components: (1) glazing, which may be one or more sheets of glass or diathermaneous material (2) absorber panel, the main function of which is to absorb the sun radiation, and to transfer it to the working medium or transfer fluid. Most solar collectors

differ mainly in the design of absorber panel it self. Some common configurations are shown in Fig. 1.2, (3) insulation, which minimizes side and downward heat loss from the absorber panel and (4) container or casing, which surrounds the above components and keeps them free from dust, moisture etc.

Flat plate collector is the simplest and most effective means of collecting solar energy, which operates in a fixed position, depending upon the design criteria. It utilizes both direct and diffuse solar radiation satisfying the major objective to collect as much energy as possible. They have been in use to heat water, water plus antifreeze additive such as ethylene glycol, water plus ammonia or other refrigerants, fluorinated hydrocarbons, air and other gases in systems like house heating and cooling, domestic and industrial water heating systems, swimming pool heating, dryers, absorption refrigeration and other systems that require thermal energy at comparatively low temperatures. Figure 1.3 explains the importance of such collectors in some of the experimented systems, (4,5,6).

Careful observation of Fig. 1.3 points out some important features of a flat plate collector system, as mentioned below.

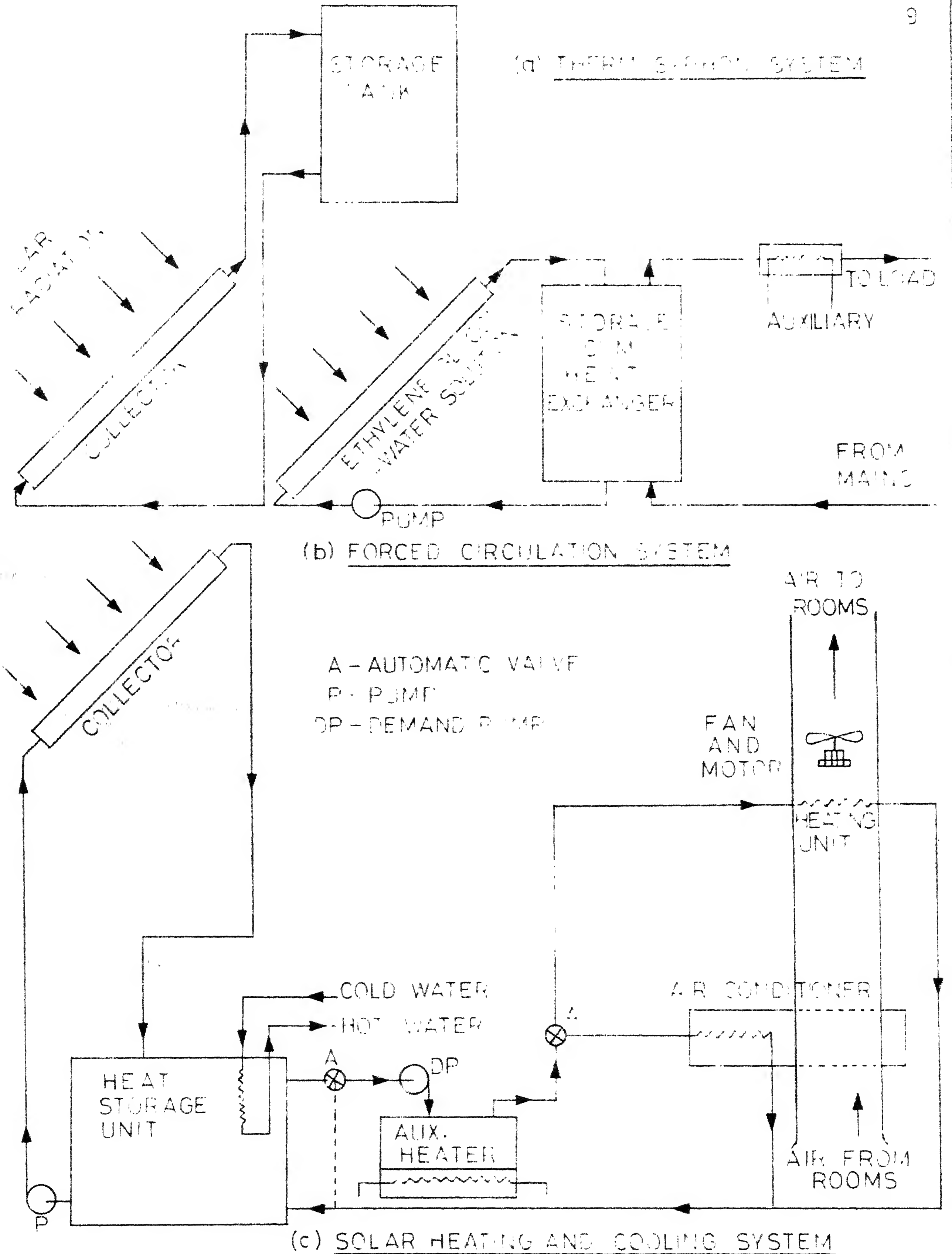


FIG.1.3 VARIOUS SYSTEMS EMPLOYING PLANE COLLECTORS

(i) Auxiliary Heat Sources:

It is realized that in temperate climates the periods of self-sufficiency in sun energy are relatively short. Also, in many applications temperature requirements of the working fluid may be quite high which may not be attainable using only the available sun energy. For successful development of a solar device in such situations, the sun energy must be regarded as a preheater source and an auxiliary heat source must be incorporated in the system to achieve the required temperature, (7).

(ii) Use of Water as Transfer Fluid:

Water has been extensively in use as heat removing medium because it is inexpensive, easily available, has good heat transfer characteristics (highest heat capacity per kg or per litre) and is suitable for thermal storage as compared to air and many chemicals.

(iii) Methods of Circulation:

Two distinct methods of circulation of transfer fluid are used - the thermosyphon and the forced circulation.

The circulation commonly called thermosyphon, occurs because of the variation of fluid density with its temperature. Fluid in the collector gets heated, its density decreases and rises upward in the absorber

tubes and is replaced by colder fluid. Process continues as long as solar radiation is strong enough to keep the collector plate sufficiently hot. Thus, circulation is without any mechanical device, but number of precautions are necessary in the use of such a simple device. Reverse flow, downwards from the tank in to the heater, there by causing loss of heat, can occur on cold nights unless tank is mounted at least 60 to 70 cms. above the top of the collector. Also, a continuous upward grade must be maintained in order to avoid the possibility of formation of air pockets in connecting tubes/pipes, otherwise circulation may cease.

It may not be possible for all the systems, to operate within the limitations mentioned above. The large storage capacity systems make it impractical to mount the tanks above the absorbers, (8). There may be situations where architectural or other considerations require that storage tank be below or at a considerable distance from the collectors, (3). In such cases, forced circulation systems are used, where recirculation of transfer fluid is achieved by means of a pump. The operation by forced circulation can take full advantage of intermittent sunshine periods and also of a lower altitude of the sun.

1.5 REVIEW OF PREVIOUS WORK

The fundamental and systematic study on the performance of flat plate solar heat collectors has been carried out earlier by Hottel and Woertz (9) with fourteen collectors, each having absorber area of 2.5 m^2. Absorber surface used was blackened copper sheet 0.5 mm thick to which were soldered six parallel copper tubes, 9.5 mm o.d., spaced 15 cms. apart, running lengthwise of the collector as shown in Fig. 1.2 (a). Collectors were having three glass covers with 25 mm air spaces and storage capacity of 17400 litres of water. Collectors were tested under forced flow conditions with a centrifugal pump and ascertained collection efficiency was 48 percent.

Czarnecki (10) performed an experiment with seven absorbers installed at various C.S.I.R.O. Laboratories, Australia, each having a tank capacity of 318 litres, with a built-in electric booster and two absorbers of total active area of 4.18 m^2, operating on thermosyphon principle. Collector construction was similar as shown in Fig. 1.2 (b). Approximately, 204 litres of water at a temperature of about 57 °C was discharged each morning. The mean yearly contribution of solar energy in the above system was reported to be 60 to 80 percent of the total energy required, depending upon the location.

Close (11) has analysed the performance with no-drawoff during the day, of a natural circulation type solar water heater with copper tubes and plate as shown in Fig. 1.2 (b). Two installations, one with 1.62 m^2 area with one glass cover and second with 1.57 m^2 area and two glass covers, inclined at 32.5^o towards north, having tank capacity of 136 litres, were experimented and the reported efficiency was 37.8 to 42.4 percent.

Yellot and Sobotka (12) have investigated the performance of a solar water heater of the type shown in Fig. 1.2 (c). The steel pipes were simply pressed-in, for thermal contact with the heat absorbing plate having 2.8 m^2 area and two glass covers. The plate had radiation absorptance as high as 0.90 and longwave emittance as low as 0.15. On a clear day, (Oct. 28, 1962) the heater, inclined at 38^o to the horizontal, operating on thermosyphon principle, gave the day long efficiency of collection of about 38.4 percent. Another aspect of the investigation by Yellot and Sobotka was to study the comparative performance of solar water heater before and after application of heat conducting cement which improved the thermal contact between the plate and the pipes. At flow rates of 44.5 and 84 kg/hr (June 7, 1963), the collection efficiency was 44.3 and 45.5 percent respectively, before using heat conducting cement. After applying heat conducting cement, the collection efficiency, for

flow rates of 86 and 85 kg/hr (June 12, 1963) was found to be 54 and 56.2 percent, respectively.

Garg and Gupta (13) have discussed the design of flat plate collector of the type shown in Fig. 1.2 (d). The collector configuration of tube-in-plate type, using various indigenous material, is optimized for maximum efficiency per unit of cost. The collector with maximum collection efficiency was found to consist of 19 mm dia. G.I. pipes at 10 cms spacing from centre to centre, bonded to a 0.5 mm thick aluminium plate.

Another type of solar water heater shown in Fig. 1.2 (e) was described by Khanna (14). It consists of a corrugated metal sheet as the absorber plate backed by a plane metal sheet to form parallel water channels running the entire length of the corrugated sheet. The corrugated sheet and the plane sheet are joined together at many points by rivets. It is reported that 450 litres of water could be maintained with a rise of 28 °C for a period of 6 hours, with absorber area of 5 m^2.

Similar kind of heater has been experimented by Rao and Suri (15). For a collector surface area of 2.32 m^2 and the tank capacity of 136 litres, the rise in temperature of water in winter at Roorkee, for natural circulation system, was from 6 °C to 50 °C (max.). The system efficiency was found to be 63 percent.

Experiments on built-in storage water heater have been reported by Garg and Krishnan (16) Fig. 1.2 (f). The heater consists of a G.I. rectangular tank of dimensions 112 cms x 80 cms x 10 cms with a capacity of about 90 litres. This tank is placed in a rectangular mild sheet tray and insulated from sides and back. Water temperature of 55 °C has been reported, for winter season with 75 perdent efficiency. Chauhan (17) has also tested similar kind of storage-cum-water heater during summer for both forced and natural circulation conditions. An average collection efficiency about 58 to 65 percent was obtained. The efficiency was observed to be nearly the same when water draw-off was taking place at 50 °C - 60 °C.

Flat plate collectors have also been used for large scale applications. Garg (4) has reported heating of 600 litres of water up to 55 °C in the winter afternoon and 48 °C to 50 °C in the early mornings at Roorkee. Circulation was provided by a pump, consuming only 0.23 kwh of electrical energy per day. System was equipped with auxiliary electric heater. The overall efficiency of the system is reported to be 50 percent.

Farber and Prescott (18) operated an engine with flat plate collector, using low boiling point fluids like trichloro-monofluoro-methane having operating temperatures between 27 °C and 72 °C.

Lof and Tybout (5) have made extensive analysis of solar heating and cooling using flat plate collectors and found that a combined system of heating and cooling was more economical than heating alone. System includes lithium-bromide-water type absorption process.

With the wide use of flat plate collectors, further developments and modifications in the design of collectors, are in progress. A solar water heater combining collection and storage has been tested in Ceylon by Chinnappa and Gnangalingam (19). The heater consists of a square coil of 7.5 cm dia. pipe (painted black to absorb solar energy), 13.5 m in length, encased in wooden box with insulation at the bottom and two glass covers. The glass surface is 1.86 m^2 in area. It was possible to get 115 to 150 litres of water at 50 °C per day, with intermittent draw-offs. Collection efficiency of 46 percent was reported.

Minardi and Chaung (20) designed and tested a proto type unit which has black liquid as a transfer fluid. Collector consists of transparent channels through which black liquid flows and directly absorbs solar energy. Base liquid used in the experiments were ethylene glycol-water, ethylene-glycol-water-India ink etc.

Girardier and Masson (21) have developed a low temperature engine which uses flat plate collector, with

an output temperature of 75 °C and a condenser temperature of 30 °C. 72 m^2 of area provides 10 m^3/hr of water for six hours a day, from a depth of 20 meters. The efficiency of the pump was reported to be very low.

1.6 THE PRESENT WORK

As described above, in most of the forced circulation systems, transfer fluid is being circulated in tubes, which are in thermal contact with the blackened absorber. Tubes are usually separated by some distance, causing the temperature of the black plate between two adjacent tubes, several degrees higher than the temperature of the tubes. Thus, the temperature distribution on the surface of the absorber oscillates between a maximum and a minimum along the number of tubes used. Since, energy losses from the absorber will depend on the difference of temperature between the absorber and the surrounding air, more losses take place from points of higher temperature difference and the efficiency of the collector may, therefore, be expected to be lower than that of a collector which has no tubes. Effective thermal bond between the plate and tubes is also difficult to achieve. Also, extensive use of tubes results in the high cost of the unit.

On the other hand, the built-in storage-type water heaters, so far studied for the forced flow situation, have got the following drawbacks:

1. bulging and leaking problems due to high water pressure necessitates complicated and costly design of the collector,
2. quantity of water coming in contact per unit area of the collector plate being quite high, decreases the efficiency of the system in terms of the maximum temperature gain.

The present work has been taken up as an attempt to overcome the problems encountered in the collector designs so far, studied. The work comprises of:

1. developing a new design of the collector of the "spray-type",
2. studying the most suitable arrangement of spray-nozzles in the above collector.

The unique feature of the spray-type collectors is that the working liquid is being sprayed upon the bottom side of the absorber plate by means of spray nozzles. The specific conical internal structure of the spray nozzle, imparts forced liquid a whirling motion which forms a spray of fine particles at the outlet. These liquid particles come in direct contact with the absorbing plate, trickle down through its entire surface and the liquid is collected at the bottom of the absorber

panel by means of gravity. A very thin boundary layer of the liquid is developed, enabling efficient heat transfer between the plate and transfer liquid.

This collector has been experimented upon, using water as heat transfer liquid, for water heating system having maximum capacity of 100 litres. The design is equally applicable for other transfer liquids used in solar energy applications. Two collectors of the same area but having different nozzle configurations have been selected to examine the performance of spray-type collectors.

Design analysis has been made to determine the absorber plate area, tank size and insulation thickness. A modified approach for determining the optimum tilt of the collector is presented. Energy balance equations for the system have been formulated and cost analysis of the collector is given to furnish an idea of its economic feasibility for large scale industrial units.

CHAPTER 2

DESIGN ANALYSIS

2.1 OPTIMUM TILT

Flat plate collector surface receives maximum energy at its surface normal to the sun rays. Consequently, the loss of intensity caused by the radiation arriving at the horizontal receiver at an angle can be greatly reduced by tilting the receiver. Ideally, the collector surface should be tilted to such an angle that it is normal to sun rays during the day. It is, however, well known that, to keep collector surface moving according to sun movement, is quite complicated and expensive an arrangement. Also the advantage to be gained by moving the flat plat collectors continuously during the day to follow the sun is not large enough to justify the installation of a moving mechanism. Because of the permanent nature of installation of collectors for systems like water and air heating, drying and refrigeration and air conditioning where very high temperatures are not required, it is absolutely necessary to select the tilt of the collector surface so as to get best possible results during the period of operation.

Various expressions have been presented in the past regarding the optimum tilt of the flat plate collector. Souka and Safwat (22) gave an expression for optimum tilt for a given solar altitude and solar azimuth

angles using normal intensity of radiation. Garg and Gupta (23) presented a relation to obtain optimum tilt for different hours and months. Kapoor and Agrawal (24) presented an expression using the apparent irradiation at zero air mass and atmosphere extinction coefficient. Besides, Kern and Harris (25) have given an expression which depends only upon mean daily direct radiation which seems impractical.

Generally, as a crude approximation, the optimum tilt has been expressed as latitude plus 15° for winter, latitude minus 15° for summer and 0.9 times the latitude for year round operation (26, 27, 28). In some cases 10° difference is also recommended. However, as tilt determines the required area of the collector which ultimately affects the cost of the unit, it should be evaluated accurately and according to the design conditions, rather than taking it as a mere function of latitude.

The present formulation* (29) takes into account the hour-durations and design periods for summer, winter and year round performance of the flat plate collectors as recommended by the India Meteorological Department (30, 31, 32). Like other authors, dependence of the tilt angle on hour angle, declination, direct, diffuse and reflected radiation, besides the latitude of the place,

*Presented at the Seventh All India Solar Energy Conference Ludhiana, Nov. 1975.

has been taken into account. Results have been compared with the existing ones and useful conclusions drawn.

2.1.1 Theoretical Analysis:

Total amount of hourly solar radiation incident upon any inclined surface can be given as, (33).

$$I_T = I_D R_D + I_d R_d + I_H R_R \qquad (2.1)$$

where,

R_D, R_d and R_R are orientation factors for direct, diffuse and reflected component of radiation, respectively.

I_D, I_d are direct and diffuse components of radiation, respectively, and I_H is total amount of radiation falling on a horizontal surface.

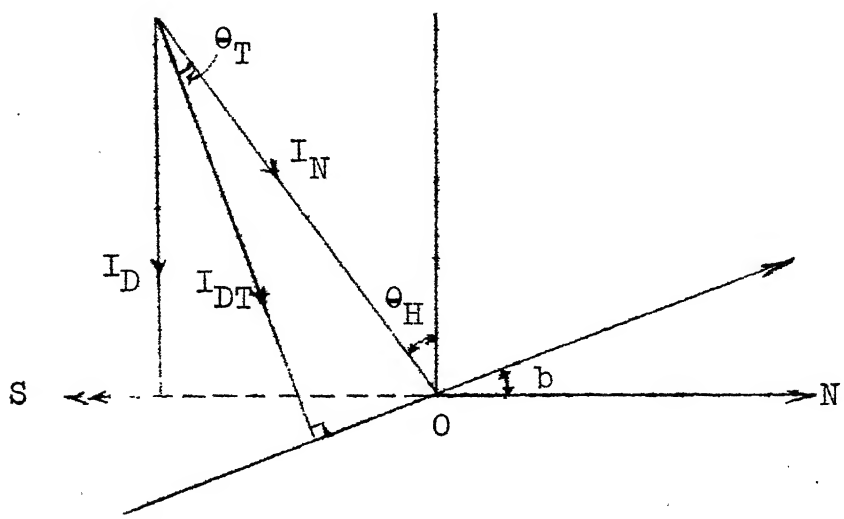


Fig. 2.1 INTENSITY DISTRIBUTION ON SURFACES

Referring Fig. 2.1, R_D can be expressed as,

$$R_D = \frac{I_{DT}}{I_D} = \frac{I_N \cos \theta_T}{I_N \cos \theta_H} = \frac{\cos \theta_T}{\cos \theta_H} \tag{2.2}$$

where,

$$\cos \theta_H = \cos L \cos h \cos d + \sin L \sin d \tag{2.3}$$

and

$$\cos \theta_T = \cos (L - b) \cos h \cos d + \sin (L - b) \sin d \tag{2.4}$$

Here, θ_T and θ_H are angles of incidence of direct radiation on an inclined surface and a horizontal surface, respectively, (2), and

L = latitude of the place

b = tilt angle of the collector with the horizontal surface

h = hour angle from solar noon

d = declination of the sun.

Assuming the diffuse sky to be isotropic i.e. uniform in all directions, and the ground having a uniform albedo, ρ, (hemispherical reflectivity), the expressions for the orientation factor for diffuse and reflected components of radiation are given by (33),

$$R_d = \frac{1}{2} (1 + \cos b) \tag{2.5}$$

$$R_R = \frac{1}{2} (1 - \cos b)\rho \tag{2.6}$$

2.1.2 Energy Absorbed by the Collector Plate :

Energy absorbed by the absorber plate mainly depends upon

(a) the insolation rate on the collector

(b) the absorptivity of the absorbing surface for solar radiation

(c) the transmittance properties of the transparent cover.

The incoming solar radiation is being transmitted by the glass cover and afterwards absorbed by the black absorbing plate.

The radiation absorbed by the absorbing plate of the inclined collector is given as

$$I_p = (\tau \alpha)_D I_D R_D + (\tau \alpha)_d I_d R_d + (\tau \alpha)_R I_H R_R \quad (2.7)$$

where,

$(\tau \alpha)_D$, $(\tau \alpha)_d$ and $(\tau \alpha)_R$ are the transmissivity - absorptivity products for direct, diffuse and reflected components of radiation, respectively.

Souka and Safwat (22) have developed a relation for $(\tau \alpha)_D$, showing transmissivity - absorptivity product as a function of incidence angle on the collector surface with the assumption that absorptivity of black paint is invariant with the angle of incidence.

$$(\tau \alpha)_D = (A - B/\cos \theta_T) \quad (2.8)$$

The value of constants A and B depends upon the kind of glass cover. Garg (23) has derived the values for A and B as 0.95 and 0.115 respectively, for a window glass commonly available in the Indian market.

For uniform sky, the values of $(\tau\alpha)_d$ and $(\tau\alpha)_R$ will be constant. From Eq. (2.7), the daily values H_T of total solar radiation absorbed by the collector plate for the number of hours j = 1 to n, on simplification, becomes

$$H_T = \sum_{j=1}^{n} I_{P_j} = \sum_{j=1}^{n} I_{D_j} \frac{(A \cos \theta_{T_j} - B)}{\cos \theta_{H_j}} + (\tau\alpha)_d \cdot I_{d_j} \left(\frac{1+\cos b}{2}\right) + (\tau\alpha)_R (I_{D_j} + I_{d_j}) \left(\frac{1-\cos b}{2}\right)\rho . \tag{2.9}$$

The hourly values of direct radiation and diffuse radiation are averaged over a particular month and if the collector unit is designed for a specific period, say winter or summer, the expression for total available solar radiation during that period becomes,

$$\sum_{i=1}^{m} H_{T_i} = \sum_{i=1}^{m} \sum_{j=1}^{n} I_{P_{ij}} = \sum_{i=1}^{m} \sum_{j=1}^{n} I_{D_{ij}} \left(\frac{A \cos \theta_{T_{ij}} - B}{\cos \theta_{H_{ij}}}\right) + (\tau\alpha)_d I_{d_{ij}} \left(\frac{1+\cos b}{2}\right) + (\tau\alpha)_R (I_{D_{ij}} + I_{d_{ij}}) \left(\frac{1-\cos b}{2}\right) \cdot \rho \tag{2.10}$$

where i indicates a particular month.

Substituting the corresponding values of various terms in Eq. (2.10), differentiating it with respect to tilt angle b, equating it to zero and solving, we get

$$b_{opt} = \arc\tan \left[\frac{\sum_{i=1}^{m} \sum_{j=1}^{n} \frac{I_{D_{ij}}}{\cos \theta_{H_{ij}}} \cdot A \cdot (\sin L \cos h_j \cos d_i - \cos L \sin d_i)}{\sum_{i=1}^{m} \sum_{j=1}^{n} (I_{D_{ij}} \cdot A + 0.5 (\tau \alpha)_d I_{d_{ij}} - 0.5 (\tau \alpha)_R \cdot \rho \cdot I_{H_{ij}})} \right] \tag{2.11}$$

Equation (2.11) is the most adequate expression for the optimum tilt of the collector for required hour - duration over the designed period of the year.

Now for uniform sky radiation, the value of $(\tau \alpha)_d$ has been reported to be 0.72 and that of $(\tau \alpha)_R$ has been considered much smaller as compared to $(\tau \alpha)_d$. Also, the albedo ρ of the ground surface is a very small quantity. Hence, neglecting the last term in the ~~denometer~~ denominator of Eq. (2.11) and simplifying, it becomes

$$b_{opt} = \arc\tan \left[\frac{\sum_{i=1}^{m} \sum_{j=1}^{n} I_{D_{ij}} \left(\frac{\sin L \cos h_j \cos d_i - \cos L \sin d_i}{\cos L \cos h_j \cos d_i + \sin L \sin d_i} \right)}{\sum_{i=1}^{m} \sum_{j=1}^{n} (I_{D_{ij}} + 0.379 I_{d_{ij}})} \right] \tag{2.12}$$

For simplicity, sun's declination angle for any hour is assumed constant throughout a month in the above analysis. Values of $I_{D_{ij}}$, $I_{d_{ij}}$ and d_i for various important Indian locations are available from standard tables, (2, 34). Values of global solar radiation and other data on maximum and minimum ambient temperatures, humidity and bright sunshine hours for Kanpur are represented in Tables 2.1 and 2.2, for our ready reference.

Equation (2.12) is programmed as per Appendix - A for solution, for various places in India for different design periods, such as summer months from March to June and Winter months from October to February, (30, 31, 32) and results for optimum tilt are represented in Table 2.3. Also, optimum tilt angles for above locations are obtained for extreme design periods, December - January in winter and April - May in summer. Results are shown in the same table 2.3.

Hour duration considered for above analysis is from 8.00 a.m. to 4.00 p.m. Effect of various hour durations upon the value of optimum tilt is shown in Table 2.4.

It can be seen from Table 2.3 that for winter operation, optimum tilt is more than the latitude of the place and for summer operation it is less than the latitude. The difference between the optimum tilt and the

TABLE 2.1 : Averages of Global Solar Radiation in Cal/cm^2.

Station Kanpur 26.4° N latitude. (Period of data November '68 to December '71).

Month/Hours	06	07	08	09	10	11	12	13	14	15	16	17	18	19	Daily Total
JANUARY		0.6	8.7	23.6	36.6	45.4	50.4	52.1	46.9	37.5	24.4	9.3	0.6		336
FEBRUARY		2.0	14.8	31.3	44.9	56.1	62.2	61.5	55.1	44.9	30.9	14.0	1.6		420
MARCH		4.8	21.2	39.8	53.4	62.7	69.8	69.2	63.6	54.4	39.1	21.8	5.8	0.2	506
APRIL	0.7	9.8	26.8	45.4	59.8	69.9	75.1	77.0	71.4	59.4	44.4	26.0	9.1	0.4	575
MAY	1.7	12.0	28.2	44.2	58.3	67.2	72.6	71.6	67.2	59.1	44.5	28.3	11.5	1.4	568
JUNE	1.9	9.8	22.9	36.5	44.7	55.1	65.8	63.1	56.7	47.7	36.6	24.1	10.9	1.3	477
JULY	1.4	9.2	21.8	36.4	45.6	56.7	59.8	55.3	54.6	45.9	37.5	24.7	10.7	1.5	461
AUGUST	0.6	8.0	19.8	30.9	39.4	47.2	47.8	47.5	47.0	44.0	32.8	21.9	7.6	0.5	395
SEPTEMBER	0.1	5.0	18.7	33.5	46.1	51.7	54.2	51.9	47.8	43.6	31.6	18.7	5.3	0.1	408
OCTOBER		2.9	16.8	34.1	49.2	59.4	63.5	62.0	59.0	47.5	33.0	17.1	3.4		448
NOVEMBER		0.9	11.0	27.2	41.2	51.6	56.5	57.4	51.7	40.4	25.9	9.9	0.7		375
DECEMBER		0.4	8.2	23.1	37.1	47.4	53.7	53.6	48.1	37.8	23.0	7.9	0.3		341

TABLE 2.2 Mean of (1) daily maximum temperature °C (2) daily minimum temperature °C, (3) relative humidity percentage and (4) mean daily duration of sunshine hours for Kanpur.

Months	(1)	(2)	(3)	(4)
JANUARY	22.8	8.6	80	8.7
FEBRUARY	26.0	11.0	69	9.2
MARCH	32.7	16.3	47	9.6
APRIL	38.3	22.0	33	9.4
MAY	41.7	27.2	35	10.2
JUNE	39.9	28.7	54	7.8
JULY	33.7	26.6	81	5.4
AUGUST	32.1	25.8	86	5.1
SEPTEMBER	32.7	24.9	81	6.7
OCTOBER	32.7	19.6	69	9.1
NOVEMBER	28.9	12.3	66	9.5
DECEMBER	24.3	8.5	78	9.2

TABLE 2.3 Optimum Tilt Angles for Various Locations in India for Various Design Periods.

Sl. No.	Place	Latitude °N	Year round	Winter* (1)	Winter* (2)	Summer** (1)	Summer** (2)
1.	Trivendrum	8.5	9.71	13.75	30.81	-2.93+	-6.17
2.	Madras	13.0	11.33	20.03	33.75	0.11	-3.26
3.	Goa	15.5	18.48	24.92	40.04	3.25	-0.95
4.	Visakhapatnam	17.75	19.22	27.79	41.05	4.82	0.74
5.	Poona	18.5	20.20	29.06	42.37	5.03	1.34
6.	Nagpur	21.1	22.43	32.24	44.71	6.89	3.64
7.	Bhavnagar	21.75	23.05	33.14	45.57	7.51	3.82
8.	Calcutta	22.65	22.33	32.90	44.71	8.07	4.73
9.	Ahmedabad	23.1	23.50	34.89	46.55	8.41	5.23
10.	Shillong	25.6	26.47	38.18	47.51	10.61	6.66
11.	Jodhpur	26.3	25.77	38.74	49.82	11.20	8.16
12.	New Delhi	28.5	27.52	41.27	51.37	12.58	9.60

* Winter months :

(1) October to February

(2) December to January

** Summer months :

(1) March to June

(2) April to May

\+ Negative tilt means collector should face North.

TABLE 2.4 Variation of Optimum Tilt with Hour Duration

Place	Poona Latitude 18.5° N optimum tilt for			New Delhi Latitude 28.5 °N Optimum tilt for		
Hour duration	Year round	Winter*	Summer**	Year round	Winter*	Summer**
7 a.m. to 5 p.m.	20.58	31.21	4.13	28.58	54.55	11.57
8 a.m. to 4 p.m.	20.20	29.06	5.03	27.52	41.27	12.58
9 a.m. to 3 p.m.	19.99	27.74	5.87	27.54	39.64	13.52
10 a.m. to 2 p.m.	19.81	26.86	6.54	27.54	38.50	14.25
11 a.m. to 1 p.m.	19.70	26.27	6.99	27.53	37.82	14.73

* Winter months, October to February

** Summer months, March to June

latitude of the place is, however not constant as stated conventionally, i.e. latitude plus 15° or minus 15° for the winter and summer months respectively. It is also observed that optimum angle changes considerably with respect to the design period. For year round operation tilt comes out to be more or less equal to the latitude of the place.

According to the meteorological data of India, the values of direct and diffuse solar radiation for many locations including Kanpur are not available. In such a situation, a rough approximation for optimum tilt may be used with respect to the latitude only. For example, the latitude of Kanpur is 26.4° N which is very close to that for Jodhpur, (Table 2.3). Hence, the optimum tilt for Kanpur is also assumed to be around 40° for five winter months (October - February) and around 50° for two extreme winter months (December - January).

Table 2.4 shows the variation of optimum tilt with various hour durations for Poona and New Delhi. It is observed that for year round operation, the optimum tilt remains nearly constant for all hour durations. For winter operation, the optimum tilt decreases as hour duration approaches noon and for summer operation, it increases as duration approaches noon.

2.2 COLLECTOR AREA

Collector plate absorbs the incident solar radiation and transfers it to the working medium. The quantity of radiation absorbed is directly proportional to the absorber plate area and, hence, careful determination of plate area becomes an important criterion in the design of flat plate collectors. Less area may not meet the design requirements resulting in low efficiency of the collector and more area may create economic and space problems.

A solar energy system operates under variable weather conditions. Exact determination of the collector area is, therefore, not possible. Czarnecki (10) and Gupta and Garg (13) have presented an approximate method to determine the area of absorber plate. Knowledge of solar radiation available to the location in question, maximum and minimum ambient temperatures, etc. on an average basis, is a prerequisite for such an analysis. For Kanpur, these data are given in Table 2.1 and 2.2, of previous section. The method to calculate the absorber area, approximately, may be explained by the following simple example:

The energy required to heat a particular quantity of fluid for a given temperature rise is expressed by

$$Q_R = M \times C_p \times \Delta T \qquad (2.13)$$

where

M = quantity of fluid to be heated

C_p = specific heat of the fluid

ΔT = required temperature rise.

Assuming a temperature rise of 50° C, desired for a particular application, the energy required per day (depending upon hours of operation) to heat 50 litres of water will be,

$$Q_R = 50 \times 1 \times 50 = 2500 \text{ kcal/day.}$$

Now the collector units, we have experimented upon, are designed for the winter months from October to February. The necessary average global solar radiation for these months available from Table 2.1 for Kanpur, is 3840 kcal/m^2/day. Assuming the efficiency of collection to be 70 percent, the net useful energy will be

$$Q_U = 3840 \times .70 = 2688 \text{ kcal/m}^2\text{/day.}$$

Thus, the required collector area is given as

$$A_c = \frac{Q_R}{Q_U} = \frac{2500}{2688} = .93 \text{ m}^2 \qquad (2.14)$$

In actual practice, however, the collector area is to be kept slightly more to take care of glass fittings, etc, so that exposed area may come out to be as calculated.

2.2.1 Specific Area of the Collector:

Generally flat plate collectors are installed indepedently in an open space, but, in many cases it may be necessary and convenient to incorporate it as a

part of the existing building either as a roof or a window shed, and integrate it with the architecture of the building. In such cases, it may not be always possible to install collector at optimum angle and correspondingly the required area will be more. This fact can be realized by determining the collector area required at various tilts. To design a collector for such situations, it may be useful to introduce the concept of "specific area" of the collector which is the area required to heat 1 litre of water by 1° C.

In the method of area calculation explained in section 2.2, the amount of useful energy Q_U is obtained from the global radiation data available on horizontal surface. If, however, both the components of radiation namely, direct and diffuse are known, the useful energy can be expressed as

$$Q_U = Q_A \times \eta_b$$

where,

Q_A = average available energy on inclined surface

$$= \frac{\sum_{i=1}^{m} H_{T_i}}{m} \qquad (2.16)$$

and η_b = efficiency of collection.

Hence, according to the definition, the specific area becomes

$$A_s = \frac{Q_R}{Q_U} = \frac{1}{Q_U} \qquad (2.17)$$

Equations (2.15, 2.16, 2.17 and 2.9) are used to calculate specific areas for two locations (New Delhi and Trivendrum) and are tabulated in Table 2.5. (Program in Appendix B).

2.2.2 Distance Between The Two Plates of Absorber Panel:

This distance mainly depends upon the geometry and specification of spray nozzles. The spray nozzles used in these collectors, give reasonably good spray at about 10 cms from its centre. Since, the height of the nozzle itself is about 2.5 cms, the distance selected is 12.5 cms.

2.3 STORAGE TANK

Although, the unit is designed for 50 litres capacity, storage tank is of 100 litres capacity in order to store more hot water and facilitate the experiment. The spherical container comprises the theoretically optimum form of accumulator with minimum surface area for the same volume and hence causes minimum heat loss. In actual practice, however, a cylindrical container having height equal to its diameter is used, to minimize heat loss. For 100 litres capacity cylindrical container, the diameter and height come out to be 50 cms. As inlet and outlet connections are on the sides of the tank, the height is taken equal to 55 cms.

TABLE 2.5 Variation of Specific Area with Tilt Angle.

Place	New Delhi		Trivendrum	
Design Period	Angle of tilt	Specific area A_s in cm^2	Angle of tilt	Specific area A_s in cm^2
Year round	16.0	5.569	2.0	5.810
	22.0	5.484	6.0	5.773
	27.52	5.459	9.71	5.762
	34.0	5.494	15.0	5.784
	40.0	5.589	24.0	5.930
Winter October to February	25.0	5.433	6.0	5.444
	36.0	5.232	10.0	5.409
	41.27	5.209	13.75	5.398
	46.0	5.228	20.0	5.428
	50.0	5.272	30.0	5.602
Summer March to June	2.0	4.933	-6.0	5.562
	8.0	4.868	-2.93	5.554
	12.58	4.853	2.0	5.573
	17.0	4.867	8.0	5.647
	22.0	4.917	14.0	5.782

Tank insulation is another important factor which has to be taken into account as large quantity of hot water has to be stored overnight with minimum temperature drop. A method, given by Schonholzer (35) is tried to determine the necessary insulation thickness as described below:

To facilitate calculation an imaginary sphere is considered, the inner surface area of which is equal to that of the cylinder, that is

$$S_i = \left(\frac{2 \pi D_i^2}{4} + \pi D_i^2 \right) = \frac{3}{2} \pi D_i^2 \qquad (2.18)$$

where, D_i is the diameter of the cylindrical tank.

The inner radius of the equivalent spherical accumulator will be

$$R_i = \left(\frac{S_i}{4\pi} \right)^{1/2} \qquad (2.19)$$

Let the insulation thickness of the required cylinder be t. The outer diameter D_o of the cylinder, therefore, will be

$$D_o = D_i + 2\,t \qquad (2.20)$$

Hence, the outer surface area of the cylinder will be

$$S_o = \left(\frac{2 \pi D_o^2}{4} + \pi D_o^2 \right) = \frac{3}{2} \pi D_o^2 \qquad (2.21)$$

Therefore, the outer radius of the equivalent spherical container will be

$$R_o = \left(\frac{S_o}{4\pi} \right)^{1/2} \qquad (2.22)$$

Assuming that the hot water is completely mixed and is at uniform temperature, the thermal capacity of the equivalent spherical accumulator is

$$W_T = V \times C_p \times w \qquad (2.23)$$

and the amount of heat stored is

$$dq = W_T \, dT \qquad (2.24)$$

where,

W_T = heat capacity of the accumulator
V = volume of water used as the heat storage medium
dT = change in temperature
C_p = specific heat of the storage medium
w = specific weight of the heat storage medium
dq = differential heat quantity.

The rate of heat loss through the walls is given by

$$-\frac{dq}{d\theta} = \frac{(T - T_a)}{R} \qquad (2.25)$$

where, θ denotes time and T_a is the ambient temperature.

The overall resistance, R, to heat transfer, is the sum of the resistances R_g and R_f through the insulating material and through the air film on the outside of the insulation, respectively.

$$R_g = \frac{1}{4\pi k} \int_{R_i}^{R_o} \frac{dr}{r^2} = \frac{1}{4\pi k} \left(\frac{1}{R_i} - \frac{1}{R_o} \right) \qquad (2.26)$$

and

$$R_f = \frac{1}{4\pi h_o} \cdot \frac{1}{R_o^2} \qquad (2.27)$$

where,

k is the conductivity of the insulating material

h_o is the convective heat transfer co-efficient.

Substituting for dq from Eq. (2.24) into Eq. (2.25) and separating variables, the following differential equation is obtained

$$\frac{dT}{(T - T_a)} = - \frac{1}{R\ W_T} \cdot d\theta \tag{2.28}$$

which upon integration, yields

$$C\ (T - T_a) = e^{-(\theta / R\ W_T)} \tag{2.29}$$

The constant of integration is obtained by using the initial condition $T = T_{\theta=0}$ at $\theta = 0$ when cooling period starts. This gives

$$C = \frac{1}{(T_{\theta=0} - T_a)} \tag{2.30}$$

The resultant temperature, T, of the storage medium after the lapse of a cooling period of time, θ, is then

$$T = (T_{\theta=0} - T_a)\ e^{-(\theta / R\ W_T)} + T_a \tag{2.31}$$

Equation (2.31) can be written in the form,

$$\left(\frac{T - T_a}{T_{\theta=0} - T_a} \right) = e^{-(\theta / R\ W_T)} \tag{2.32}$$

Guessing various values of cylinder insulation thickness ranging from 2 cms to 25 cms and using the values of conductivity $k = 0.032$ kcal/hr in ^oC and convection co-efficient $h_o = 5.0$ kcal/hr $m^2\ ^oC$, for glass wool insulation, the ratio $(T - T_a) / (T_{\theta=0} - T_a)$ and temperature

drop ($\bar{T}_{\theta=0}$ - T) after the lapse of time, θ, is calculated and stated below for the following set of conditions.

$$\bar{T}_{\theta=0} = 70\ ^{o}C;\ T_a = 10\ ^{o}C,\ \theta = 12\ \text{hrs},\ D_i = 0.5\ m,$$

Cylinder insulation thickness, cms.	Non-dimensional temperature difference ratio	Temperature drop of water during cooling period ^{o}C
2	0.876	7.44
3	0.9074	5.56
5	0.9366	3.8
10	0.96	2.4
15	0.9691	1.85
25	0.9757	1.46

From the above table, it is clear that beyond 10 cms thickness of insulation the temperature drop is not significant. We have, therefore, chosen 10 cms thickness of insulation as it is not justifiable to incur more expenditure by choosing higher value of insulation thickness.

2.4 SYSTEM ANALYSIS

A considerable amount of work has been carried out on the method to determine the thermal performance of plate solar water heating systems, (9, 11, 36, 37, 38, 39, 40). Recently Ong (41, 42) has reported a mathematical model of a natural circulation water heating

system. The present analysis is based upon the above model with basic difference in some of the assumptions, regarding forced circulation system.

Figure 2.2 represents, schematic diagram of the present forced circulation water heating system. The following assumptions are made.

(a) The entire system is split up into two sections, each section being identified by the suffix j (j = 1,2). The first section (j=1) represents the storage tank and connecting piping between tank outlet and collector inlet. The second section (j=2) represents the complete collector unit containing the glass cover, absorber plate and piping between the collector outlet and tank inlet.

(b) Both the sections are assumed uniform in temperature and denoted by section temperature T_j. Because of the direct contact between the plate and water, outlet water temperature is assumed equal to plate temperature in section 2 ($T_2 = T_p$).

(c) Inlet temperature T_1 to absorber unit is equal to the mean temperature of water in the tank (T_m) because of the forced circulation system.

(d) The mean temperature of the pipe surface and water flowing inside are equal, and so also, the mean tank surface temperature and that of water contained in it.

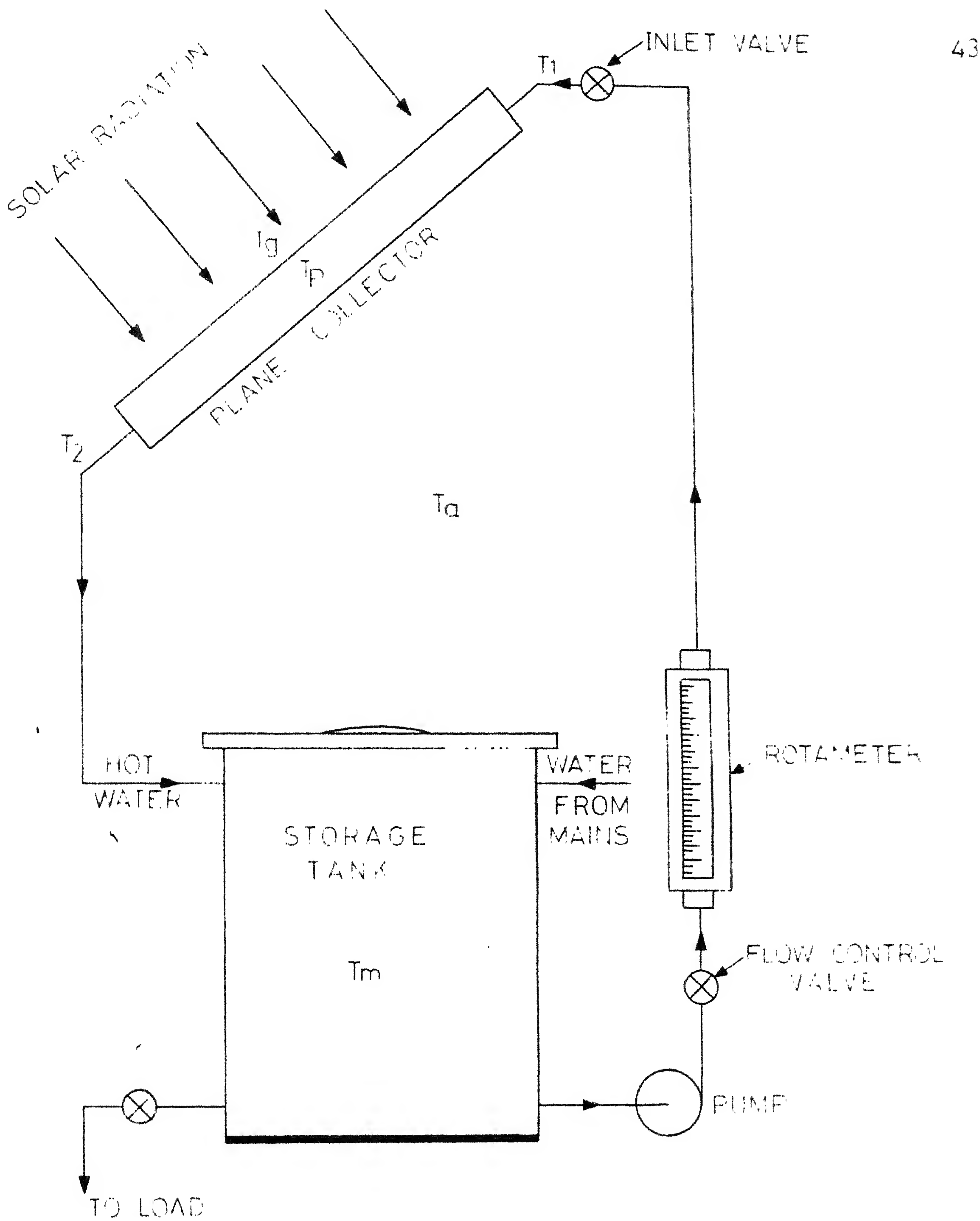


FIG. 2.2 SCHEMATIC DIAGRAM OF THE EXPERIMENTAL SET-UP

(e) The thermal capacity of each section has been taken into account rather than heat capacities of the water and the container material separately.

2.4.1 Energy Balance Equation:

The instantaneous, heat balance equation for any section, in general, can be written as,

$$Q_s = Q_a - Q_l - Q_c \tag{2.33}$$

where,

Q_s = heat stored within section = $W_j \dfrac{d\,T_j}{d\theta}$

Q_a = radiant heat absorbed = $\alpha_j^* A_j I_p$

Q_l = total heat loss = $U_j A_j (T_j - T_a)$

Q_c = net heat convected out = $\dot{m}\,(T_j - T_k)$

$k = 2$ for $j = 1$

$k = 1$ for $j = 2$

where,

α_j^* = is heat absorbing coefficient

= 1 for radiation absorbing section ($j = 2$)

= 0 for non-absorbing section.

$d\theta$ = time interval

$\dot{m}$ = mass flow rate of liquid.

U_j = overall heat loss co-efficient

T_j = mean section temperature

T_a = ambient or surrounding air temperature

W_j = thermal capacity of the section = the sum of heat capacities of water and the container material.

I_p = Intensity of radiation absorbed by the inclined plate.

Substituting the corresponding values in above Eq. (2.33) and presenting differential form in finite difference form we get,

$$W_j (T_j^* - T_j) / \Delta\theta = \alpha_j^* A_j I_p - U_j A_j (T_j - T_a) - \dot{m} (T_j - T_k) \qquad (2.34)$$

where T_j^* is the mean section temperature after a small finite increment of time $\Delta\theta$.

On rewriting Eq. (2.34),

$$T_j^* = T_j + (\Delta \theta / W_j) \left\{ \alpha_j^* A_j I_p - U_j A_j (T_j - T_a) - \dot{m} (T_j - T_k) \right\} \qquad (2.35)$$

In Eq. (2.35), value of I_p can be obtained from Eq. (2.7). Overall heat loss co-efficient for the tank U_1, is given as, (42).

$$U_1 = 0.10 (T_m - T_a)^{1.25} \qquad (2.36)$$

and overall heat loss co-efficient for collector U_2, is given as, (9)

$$U_2 = \frac{1}{\dfrac{n_g}{C_w \dfrac{(T_p - T_a)^{1/4}}{(n_g + f)^{1/4}}} + \dfrac{1}{h_w}} + \frac{\sigma (T_p^4 - T_a^4)}{\left(\dfrac{1}{e_p} + \dfrac{2n_g + f - 1}{e_g} - n_g\right)(T_p - T_a)} \qquad (2.37)$$

Equation (2.35) can be solved for mean section temperatures by substituting the values of various parameters involved. We have not been able to calculate these values, since the direct and diffuse radiation data for Kanpur are not available from the India Meteorological Centre. Also we could not measure it independently because of the non-availability of the radiation measuring instrument.

2.5 COLLECTOR AND SYSTEM EFFICIENCIES

Instantaneous efficiency η_i, defined by

$$\eta_i = \dot{m}\, C_p \,(T_2 - T_1) \,/\, I_p \, A_c \tag{2.38}$$

is a measure of the heat collection ability of the absorber unit. It indicates the rate at which incident radiation energy is being converted into useful thermal energy.

In order to evaluate the overall performance of the solar water heater system for a certain period of the day a bulk efficiency η_b is defined by,

$$\eta_b = W_T \,(T_f - T_i) \,/\, A_C \, H_T \tag{2.39}$$

which is the ratio of the increase in thermal energy stored in the tank and the sum total of radiation energy falling upon the collector plate surface for the duration considered.

CHAPTER 3

EXPERIMENTAL SYSTEM

3.1 SYSTEM DESCRIPTION AND COMPONENTS

Experimental system basically consists of four main components:

(1) Collector

(2) Storage tank

(3) Fluid circulating pump

(4) Connecting arrangement

Water has been used as heat transfer liquid and application has been limited to water heating. Circulation pump draws water from the storage tank and forces it to the inlet of the collector in which spray of water takes place. Fine particles of water come in direct contact with the bottom of the absorbing plate, and move downwards by means of gravity, adjacent to the absorber plate, which is inclined at the optimum tilt (40^{o} for Kanpur for five winter months). Heated water from the collector outlet returns to the tank again by means of gravity. The tank is located at the convenient position lower than the collector. Figure 3.1 shows the photograph of the experimental set-up. The details of each component and its functions are described below:

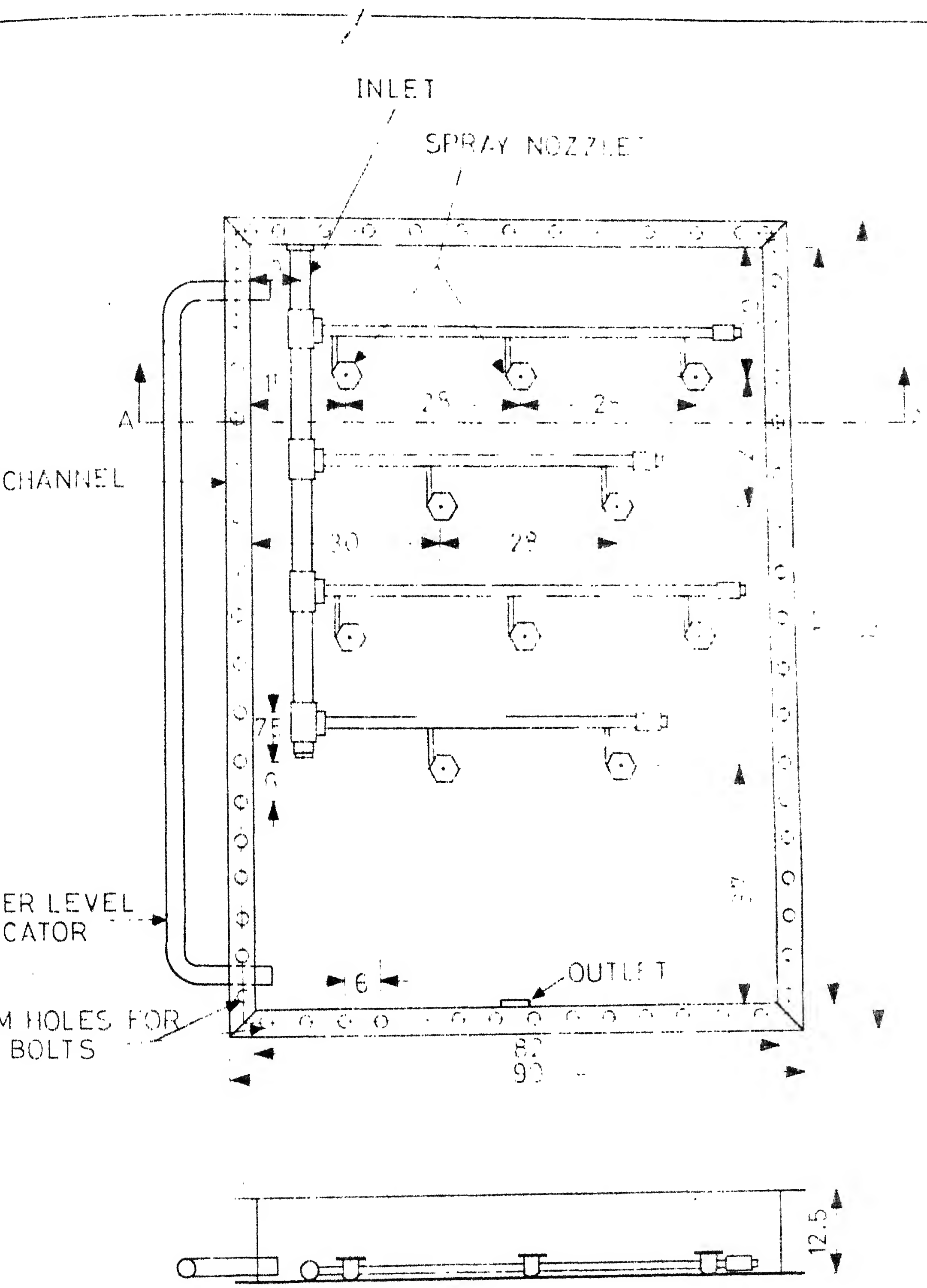
(1) Collector:

This is the most important component of the system which receives the incoming solar radiation, collects it and transfers it to the working fluid. It is comprised of four parts.

(i) Absorber panel
(ii) Glass cover
(iii) Insulation
(iv) Casing

(i) Absorber Panel

The panel consists of a black metal plate at the top, a bottom plate, metalcasing and nozzle structure resting upon the base plate. Top plate is painted with dull absorbing black paint mixed with black mesh powder in order to increase its absorptivity. Top plate is 22 gauge G.I. sheet of size 125 cms x 90 cms and bottom plate is 20 gauge G.I. sheet having the same area. The distance between the two plates is 12.5 cms. Bottom plate has been soldered to the metal casing, forming a base for nozzle structure. Upper absorbing plate has been bolted to the casing channel with 8 mm (5/16") G.I. bolts and nuts so that, if required, it can be removed easily. Number of



SECTION A-A

ALL DIMENSIONS IN CMS

FIG. 3.2 NOZZLE CONFIGURATION OF COLLECTOR-I

(1") dia. inlet and outlet. Casing channel consists of 12 mm (1/2") dia. polythene tube, at its outer side, to be used as water level indicator.

Spray system consists of nozzles and piping. Collector-I has 10 spray nozzles fixed in 9.5 mm (3/8") dia. sockets, welded to 12 mm (1/2") dia. G.I. pipes. These pipes are connected to 25 mm (1") dia. inlet pipe. Figure 3.2 shows the details of the nozzle configuration having 3-2-3-2 combination.

Collector-II differs from I in the sense that it consists of only 5 spray nozzles, all on the same axis connected to the inlet pipe similar to that in collector I (Fig. 3.3). Outlet has been increased* to 32 mm ($1\frac{1}{4}$") dia. Also, the distance between the top end and the nozzles has been reduced in order to increase the contact area of the absorber plate.

Spray nozzles are made of high temperature - resistant plastic with specific internal conical shape which imparts whirling motion to the forced fluid and a spray of fine particles is available at the nozzle outlet.

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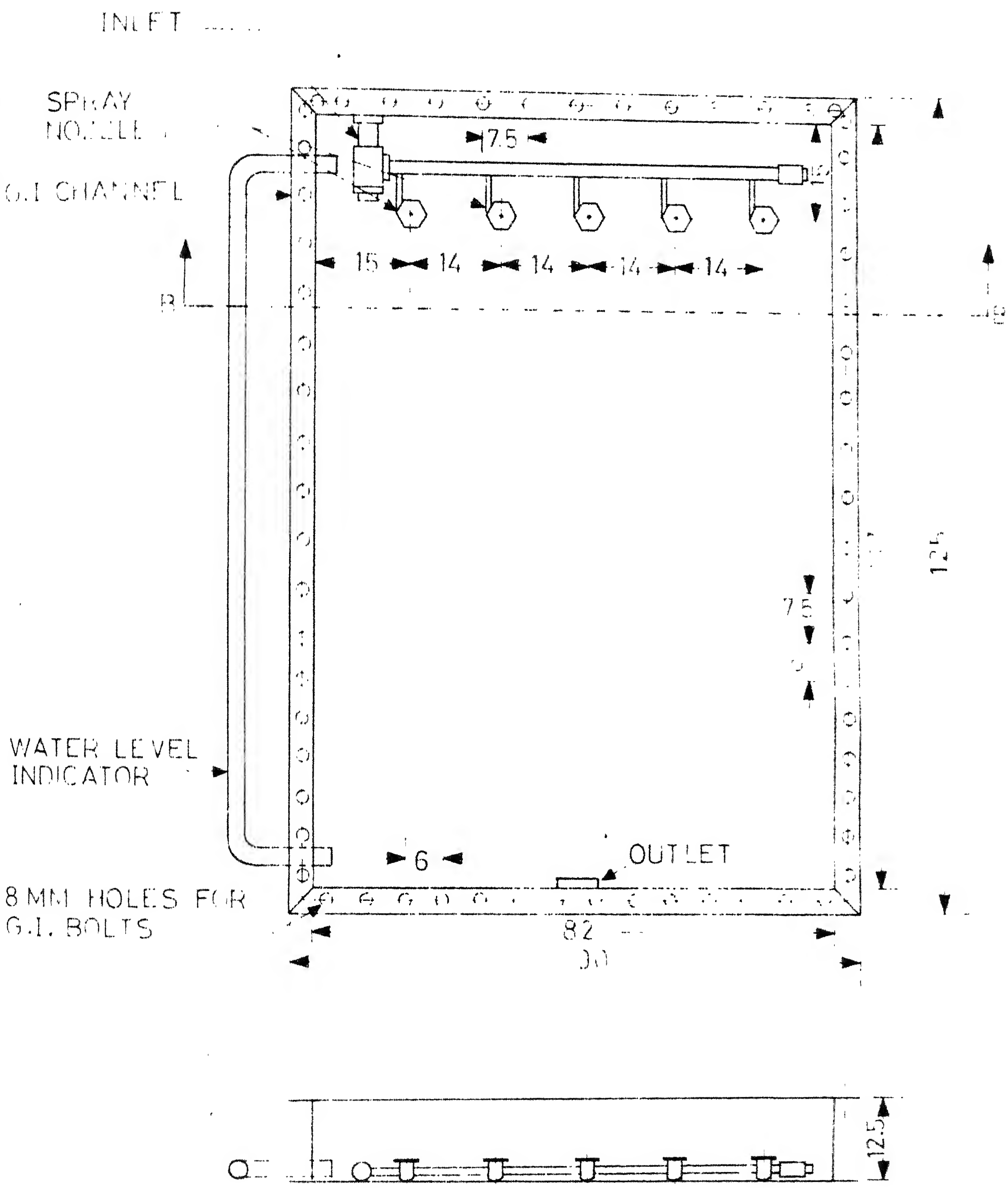
*During experimentation of Collector-I, it was observed that 25 mm (1") dia outlet restricts the flow rate to only 3 gpm beyond which, the water starts accumulating inside the collector. This difficulty is overcome in Collector-II by increasing the outlet to 32 mm ($1\frac{1}{4}$") diameter.



ALL DIMENSIONS IN CMS.

FIG. 3.3 NOZZLE ARRANGEMENT FOR COLLECTOR-II

Figure 3.4(a) shows the details of a spray nozzle.

(ii) Glass Cover :

Glass has been the principal material used to glaze solar collector mainly because of its high transmissivity of incoming shortwave solar radiation. Besides, it is invarient to the sunlight and weather conditions. Glass is opaque to the longwave radiation emitted by the absorber plate and, hence, it acts as an effective heat trap when used as a cover. It also reduces losses by convection and radiation, as it forms a stagnant air gap above the absorber plate. A common window glass 4 mm thick, has been used to cover the absorber panel yielding a net exposed area of 0.98 m^2. The stagnant air gap between the glass cover and the absorber is taken as 3.5 cms. The glass is held in a recess in the wooden box by gaskets and wooden strips which permit thermal expansion but prevent the entrance of dust and moisture. Dust must be excluded since it coats the collector plate, which reduces its absorptance.

(iii) Insulation :

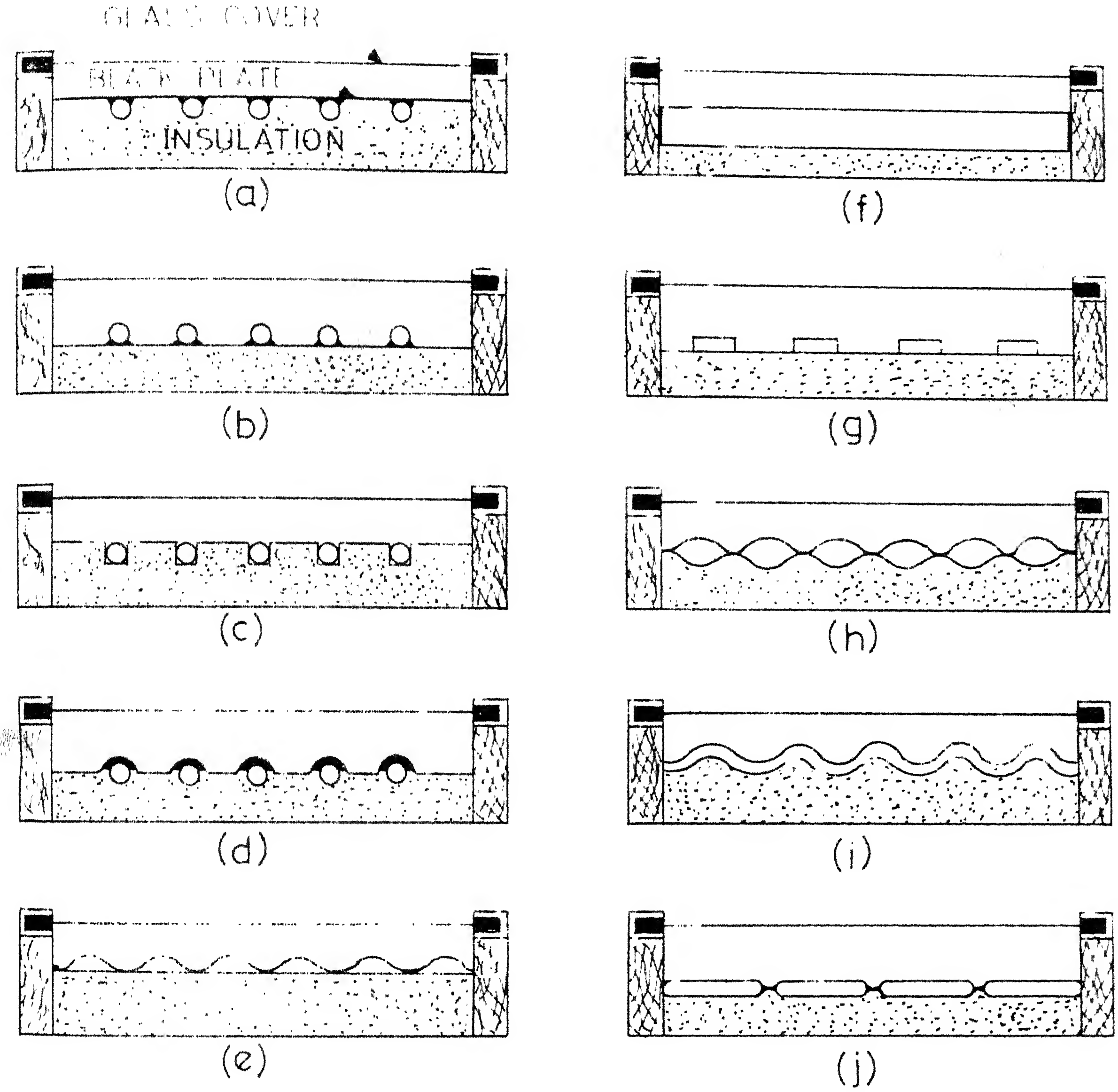
In order to prevent heat losses from the sides and bottom of the absorber panel, a 4 cms thick layer of glass-wool is held, between the absorber panel and the wooden casing. Specific shape of the absorber channel provides necessary space for side insulation.

(iv) Casing :

The abosrber panel and insulation are enclosed in a wooden box (made of Deodar wood) which holds it securely. The overall dimensions of the wooden box are 130 cms x 95 cms x 25 cms. Wooden casing consists of two parts - upper part which holds glass cover and lower part which contains absorber panel with rubber gasket in between, to prevent air circulation. Two-parts enclosure design makes it convenient to remove the absorber panel and insulation, if necessary, without disturbing the glass cover. Figure 3.4 (b) shows the cross-section of the complete unit with an enlarged portion of one of the ends of the collector unit.

(2) Storage Tank :

A cylindrical storage tank of 20 gauge G.I. sheet having 50 cms dia. and 55 cms height, stores about 100 litres of water. It has 25 mm (1") dia. inlet at the top, and 12 mm (1/2") dia. supply and load distribution connections. Tank is insulated by 10 cms thick glass-wool layer from all sides in order to prevent heat losses. Insulation has been covered by polythene sheets to protect it against weather effects. Hot water from the collector enters the tank from top inlet, mixes with the tank water and again drawn by the pump from the bottom inlet for recirculation.



(a) TUBES BONDED TO LOWER SURFACE OF BLACK PLATE
(b) TUBES BONDED TO UPPER SURFACE OF THE PLATE
(c) TUBES PRESSED IN TO THE PLATE
(d) TUBES BONDED TO BOTTOM OF THE CURVED PLATE
(e) CORRUGATED SHEET ATTACHED TO FLAT PLATE
(f) BOX TYPE STORAGE-CUM-HEATER
(g) RECTANGULAR TUBES BONDED TO PLATE
(h) CORRUGATED PLATES FASTENED TOGETHER
(i) CORRUGATED SHEETS FORMING CURVED PATH
(j) FLAT PLATES DIMPLED AND SPOT WELDED

.1.2 VARIOUS DESIGNS OF ABSORBER PANELS

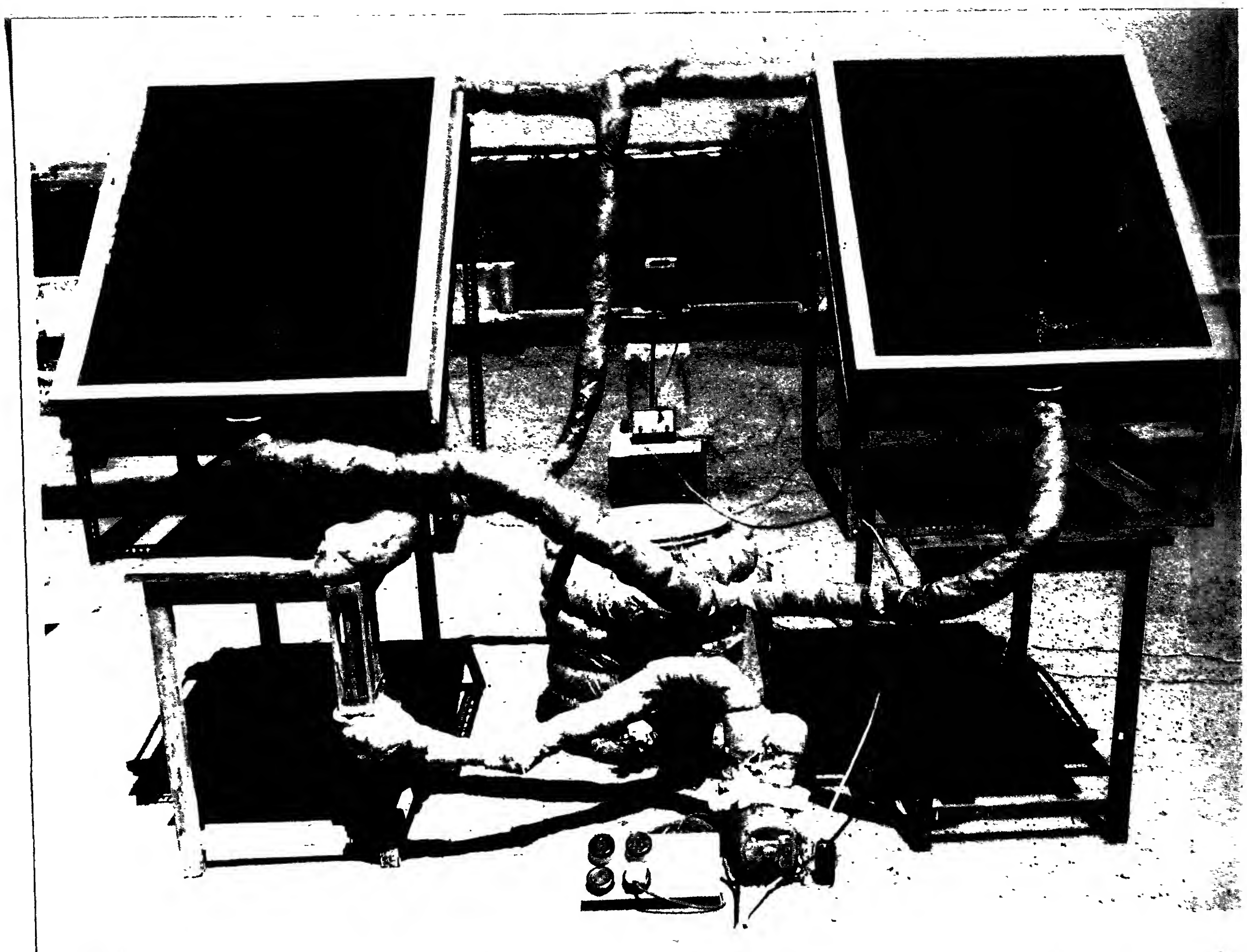


FIG. 3.1 A PHOTOGRAPHIC VIEW OF THE EXPERIMENTAL SET-UP

(3) Fluid Circulating Pump:

Water circulating pump with 0.5 hp has been used in the experiment. This pump is capable to give flow rate up to 7 to 8 GPM which is beyond the requirement. Low hp pump can also serve the purpose easily.

(4) Connecting Arrangement :

Various components, mentioned above, are joined by flexible polythene tubes for the experimental purpose. Tank outlet and pump are connected by 25 mm (1") dia. tube, pump outlet and inlet are connected with 12 mm (1") dia. tube and collector outlet and tank inlet are joined by 25 mm (1") dia. rubber hose in collector-I and 32 mm (1 $\frac{1}{4}$") dia. rubber hose in collector-II. All the tubes are insulated by 4 cms thick glass wool to minimize the heat losses.

3.2 INSTRUMENTATION

(a) Temperature Measurements:

All temperatures of the collector unit and storage tank have been measured by calibrated copper-constantan (24 gauge wire) teflon insulated thermocouples. Three thermocouples are soldered to the black absorber plate, two to back plate and one to the glass cover. Two thermocouples have been soldered to the tank surface. Average tank water temperature has been measured by thermocouples which remained dipped in the water.

Ambient temperature, collector outlet and inlet water temperatures have been recorded by thermometers placed in shade near the tank.

(b) Wind Direction and Velocity Measurement

A windscope has been used to measure both the direction and velocity of the wind. The windscope has a vane and revolving cups mounted in suitable positions. Depending on the direction of the vane and speed of the revolving cups, indications are obtained on an instrument connected through a cable to windscope.

(c) Flow Rate Measurement

Flow rate of the water circulation has been measured by Brooks Rotameter ranging between 0.5 GPM and 5.0 GPM. Flow rate is controlled by means of a control valve placed just after the pump on outlet side. Rotameter is mounted between pump outlet and collector inlet.

(d) Radiation Measurement

In the absence of the availability of radiation measuring instruments in our laboratory, the radiation data has been obtained directly from the Central Laboratory, National Radiation Centre, Poona for Kanpur and other important locations in India. This data have been used to determine the collector area for Kanpur and optimum tilt angles for many Indian locations as stated in the previous chapter.

CHAPTER 4

RESULTS AND DISCUSSION

4.1 TEST PROCEDURE

Spray-type collectors designed as described in Chapters 2 and 3, have been experimented under normal weather conditions at Kanpur, for the month of March, from March 2, 1976 to March 21, 1976. Collector-I is tested for seven days, (March 2 to March 8), for flow rates ranging from 2 GPM to 3 GPM, and Collector-II for 8 days, (March 13 to March 21) for flow rates ranging from 1 GPM to 4 GPM for 50 litres and 100 litres of water, as the system load. Operation with lower or higher flow rates than mentioned above, results in inadequate spray or accumulation of water inside the absorber panel, as the case may be.

A specific test has been conducted in which overnight stored hot water is recirculated next day. Also, on March 7 experiment is carried out in which system is being evacuated after reaching 62 °C and recharged for the operation during afternoon hours.

On an average, the system has been operated continuously for 6 to 8 hours a day. Each morning fresh water has been charged in the system and heated water is stored overnight in the insulated storage tank to study the storage characteristics.

4.2 ANALYSIS OF RESULTS

Experimental results are presented in Figures 4.1 to 4.5 for Collector-I and Figures 4.6 to 4.13 for Collector-II. These figures represent the behaviour of mean absorber plate temperature, mean tank water temperature, ambient temperature and temperature rise of the water in collector (difference between the outlet and inlet of the collector), on an hourly basis, for various flow rates and system capacities. Each figure conforms for a single day observations. The figures depict the following facts:

(1) In the evening hours of the day, the mean tank water temperature approaches the mean absorber temperature. This is due to the fact that the amount of incident radiation during evening hours decreases and hence the plate temperature stops increasing. The water temperature, however, keeps on gaining heat from the absorber plate till it reaches the maximum equilibrium temperature. Further recirculation of water results in the decrease of water temperature as shown in the figures.

(2) During morning hours, the temperature of tank water is low. The temperature difference between the incoming water and the absorber plate being high, the rate of rise of water temperature is quite significant during morning hours as compared to evening

hours. However, in Fig. 4.1, the behaviour of the plate water temperatures does not comply to the above. This is because the day was very windy and cold and the experiment was commenced much early (7.30 a.m.) without allowing the plate to gain higher temperature.

(3) The plot of temperature rise of water in collector shows that it attains a maximum temperature around noon. This is because the intensity of incoming radiation is highest during this period of the day as indicated in Table 2.1 (average value of global solar radiation during March being 62.7 cal/cm^2 at 11.00 a.m. and 69.2 cal/cm^2 at 1.00 p.m.).

(4) As the flow rate of circulation of water increases, temperature rise of water in the collector decreases and also the difference between the plate temperature and mean tank water temperature decreases. This is because in the spray type collectors, heat transfer occurs due to the direct contact between water particles and hot plate. Thus, the rate of heat removal depends upon the duration of contact. As the flow rate increases the discharge velocity of water particles from nozzles increases thereby, decreasing the contact time. Hence, the rise in water temperature is relatively low, compared to that at low flow rates. Increase in flow rate, also brings mean tank water temperature very close to the absorber plate

ate	Flow rate GPM	System load in Litres	Initial ambient temp. °C	Mean tank water temperatures °C Initial	Final	Overall rise in tank water temp. °C	Time From a.m.	To p.m.	Operation time in hours	Average rise in temp. per hour	Energy collected per day kcal/day
ollector-I											
arch 2	3	100	13.0	26.5	70.0	43.5	7.30	4.30	9.00	4.85	4350
arch 3	3	100	26.0	51.3	77.5	26.2	11.00	3.45	4.45	5.50	2620
arch 4	3	50	26.0	26.0	77.0	51.0	10.00	3.30	5.30	9.30	2550
arch 5	2	100	22.5	29.5	68.4	38.9	9.00	4.30	7.30	5.20	3890
arch 6	2	50	22.5	26.0	79.0	53.0	9.00	4.30	7.30	7.00	2650
arch 7	2.5	100	22.4	27.5	61.0	33.5	8.30	1.00	4.30	7.45	3350
arch 7	2.5	100	30.5	28.5	48.0	19.5	2.00 (p.m.)	4.45	2.45	7.10	1950
arch 8	2.5	100	22.0	34.0	71.5	37.5	9.00	4.30	7.30	5.00	3750
ollector - II											
arch 13	1	100	23.0	33.5	71.6	38.1	8.30	3.30	7.00	5.45	3810
arch 14	1	50	21.0	30.0	80.0	50.0	8.30	3.00	6.30	7.70	2500
arch 15	2	100	22.0	29.0	68.5	39.5	9.00	3.30	6.30	6.10	3950
arch 16	2	50	22.5	25.0	80.0	55.0	9.15	4.30	7.15	7.60	2750
arch 17	3	100	23.0	28.5	67.5	39.0	9.30	3.30	6.00	6.50	3900
arch 19	3	50	24.0	26.0	79.0	53.0	9.30	3.30	6.00	8.84	2650
arch 20	4	100	23.0	29.0	67.5	38.5	9.00	3.30	6.30	5.90	3850
arch 21	4	50	28.0	29.0	75.0	46.0	10.15	3.30	5.15	8.75	2300

The effects of flow rate and system load have, further, been studied by considering the overall temperature rise per hour of the operation period. This has been presented in Table 4.1 and also in Fig. 4.14. It is seen that for 50 litres of water load on the system, the average temperature rise per hour is significantly large as compared to that for 100 litres of water.

It is interesting to note that the Collector-II, which is the preferred design over Collector-I, attains the maximum average rise in tank water temperature at a flow rate of 3 GPM.

Figure 4.15 shows, the experimental results for the case of recirculation of water, heated on the previous day and stored overnight. Maximum temperature obtained in this case is 77.5 °C, allowing for an overall temperature rise of 26.2 °C in 4.45 hours of operation (11.00 a.m. to 3.45 p.m.). This experiment confirms the fact that at higher initial temperature of water, the heat gain from the collector is considerably less.

Figure 4.16 shows the test results for a typical experiment in which 100 litres of water at initial temperature of 27.5 °C is first heated to 62 °C in 4.30 hours (8.30 a.m. to 1 p.m.) of operation, collecting 3350 kcal of heat from the absorber plate. The system was then evacuated, recharged with 100 litres of fresh water at

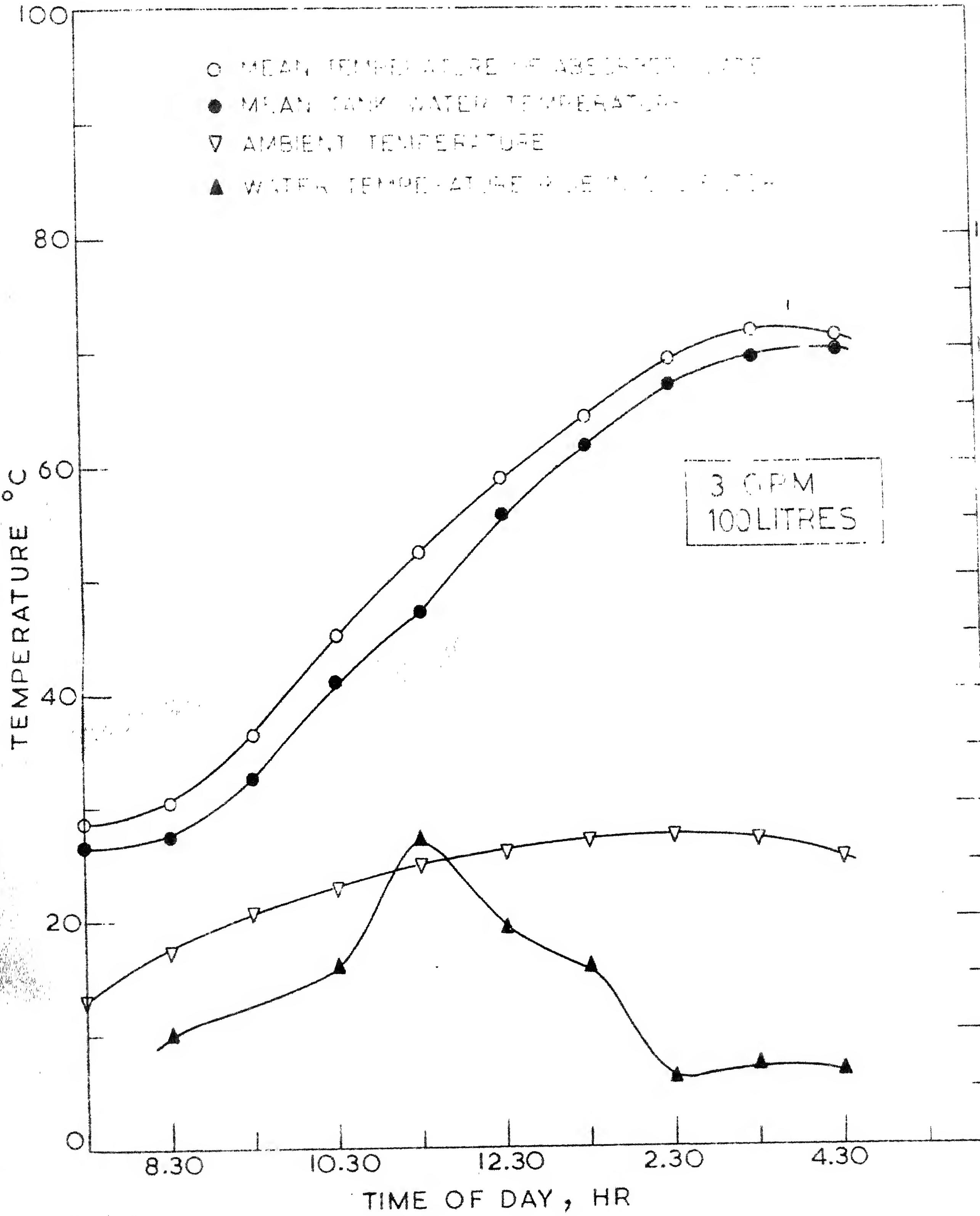


FIG. 4.1 EXPERIMENTAL RESULTS , MARCH 2 , 76.

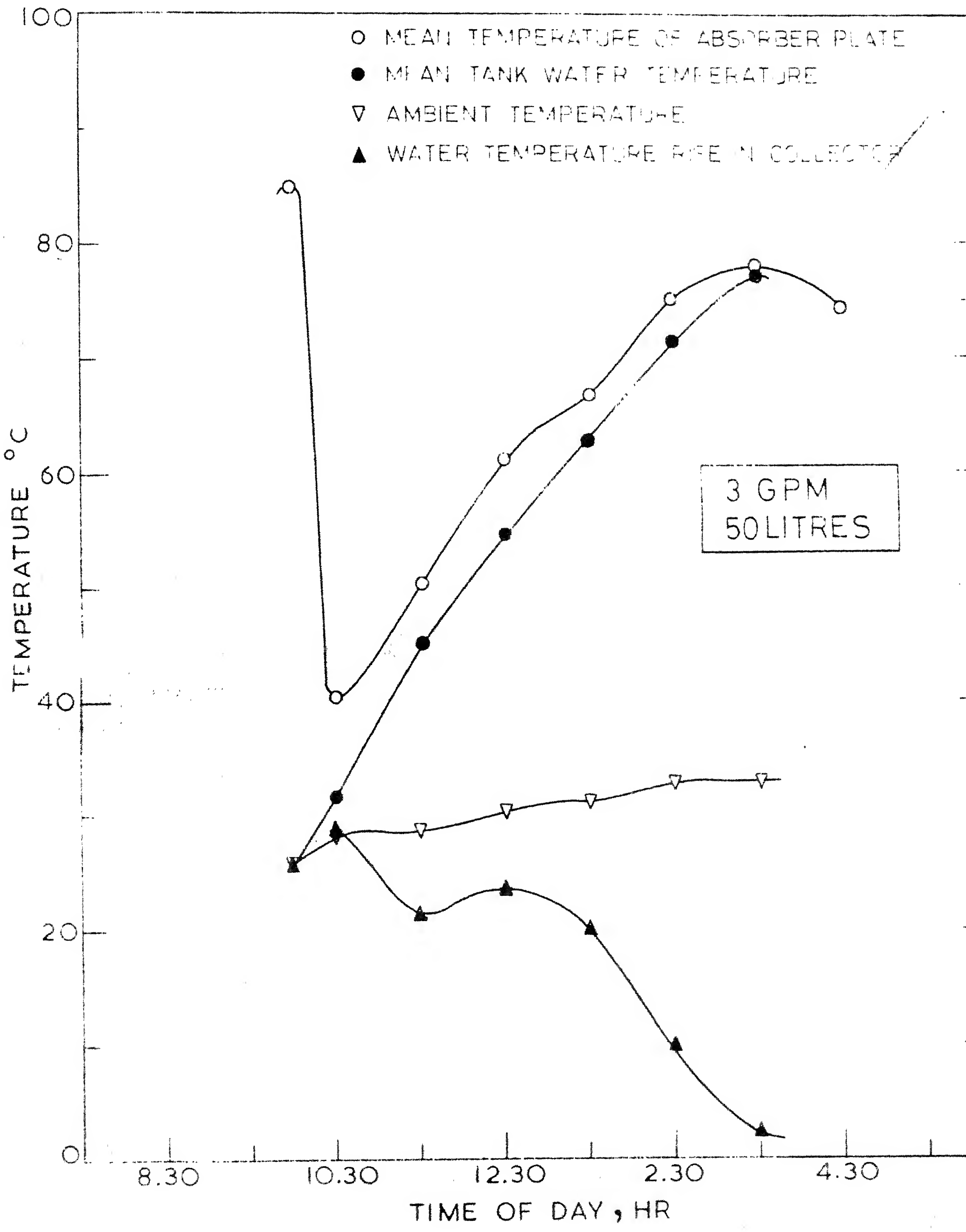


FIG. 4.2 EXPERIMENTAL RESULTS, MARCH 4, 76.

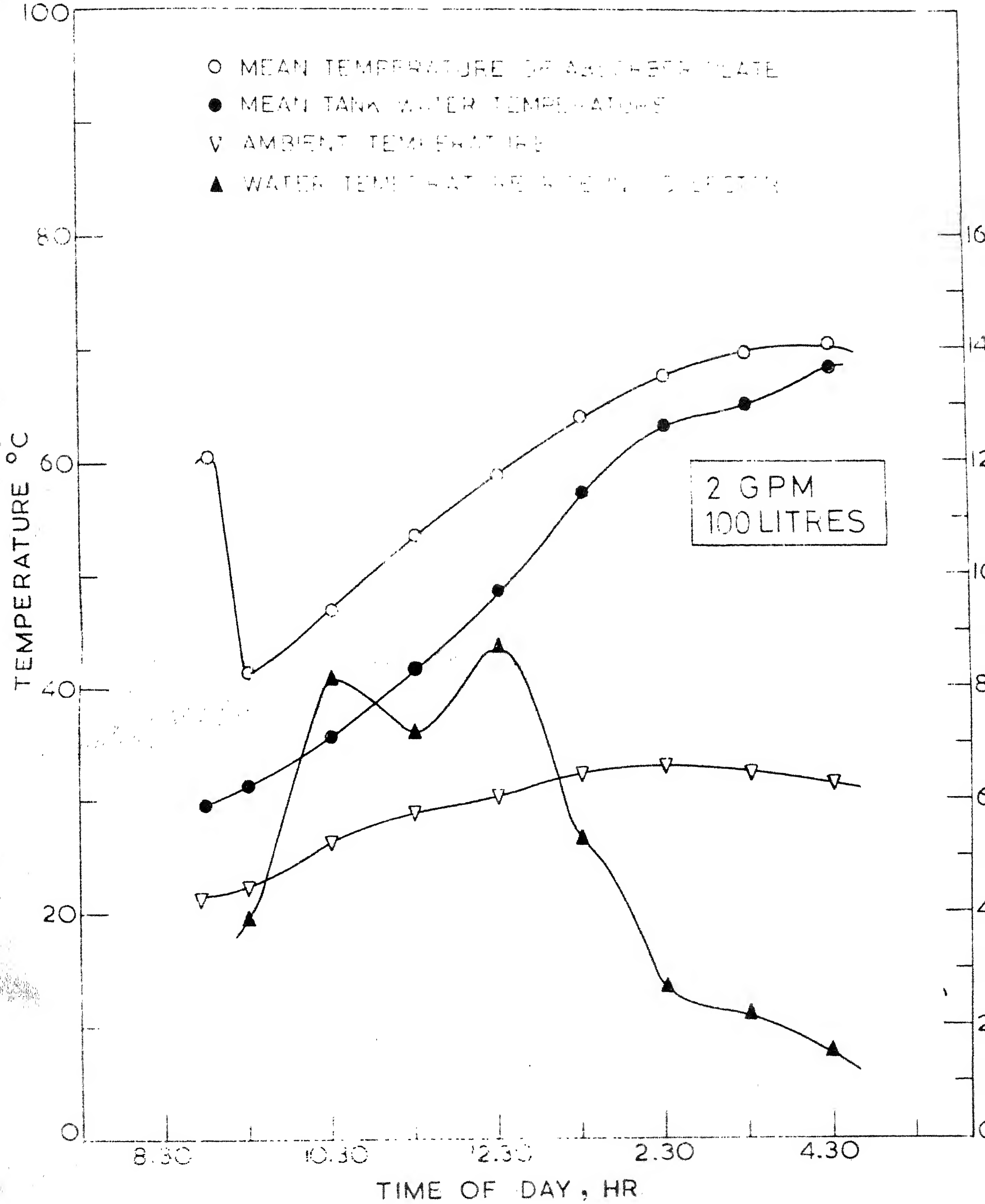


FIG. 4.3 EXPERIMENTAL RESULTS , MARCH 5 ,76.

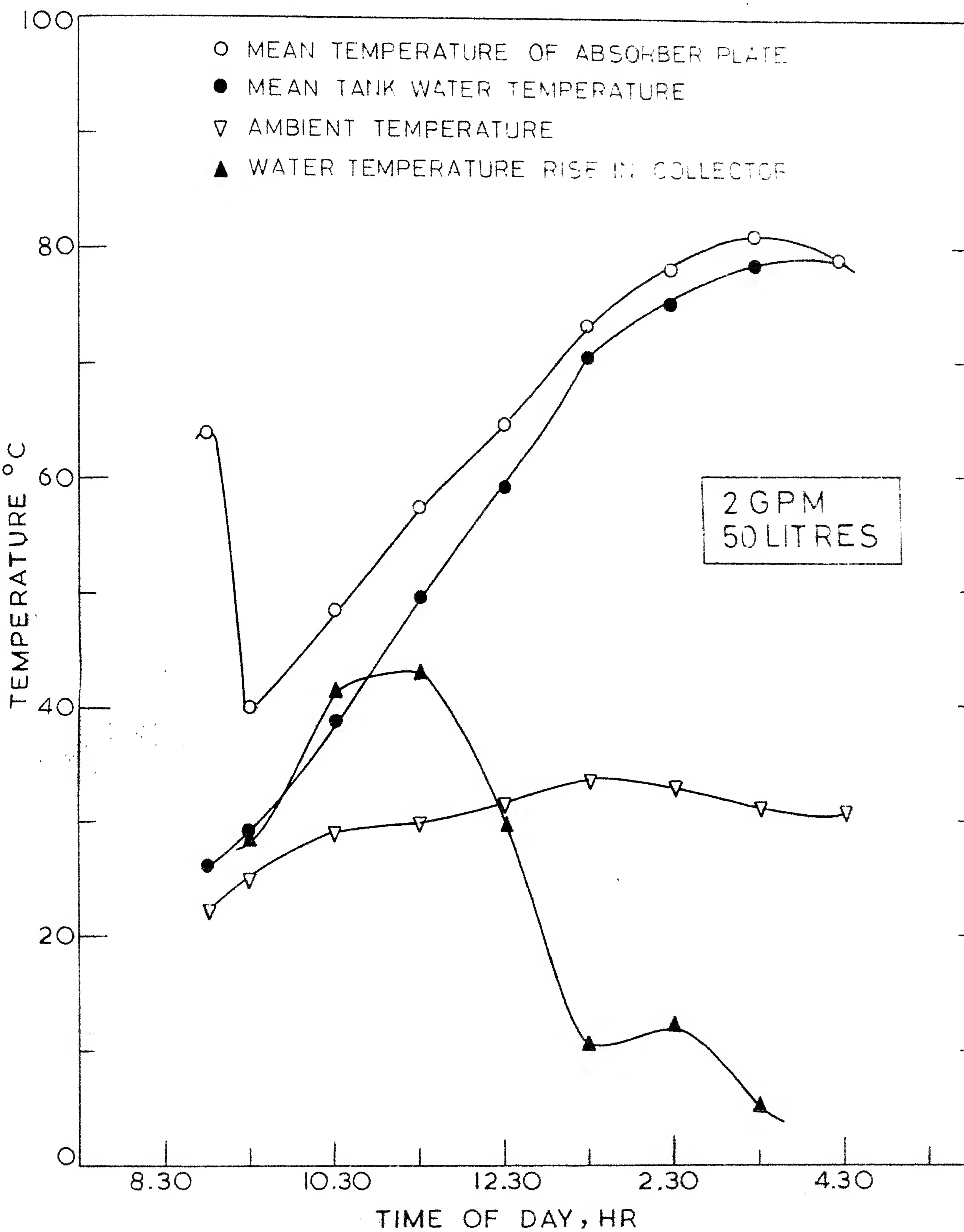


FIG. 4.4 EXPERIMENTAL RESULTS , MARCH 6 ,

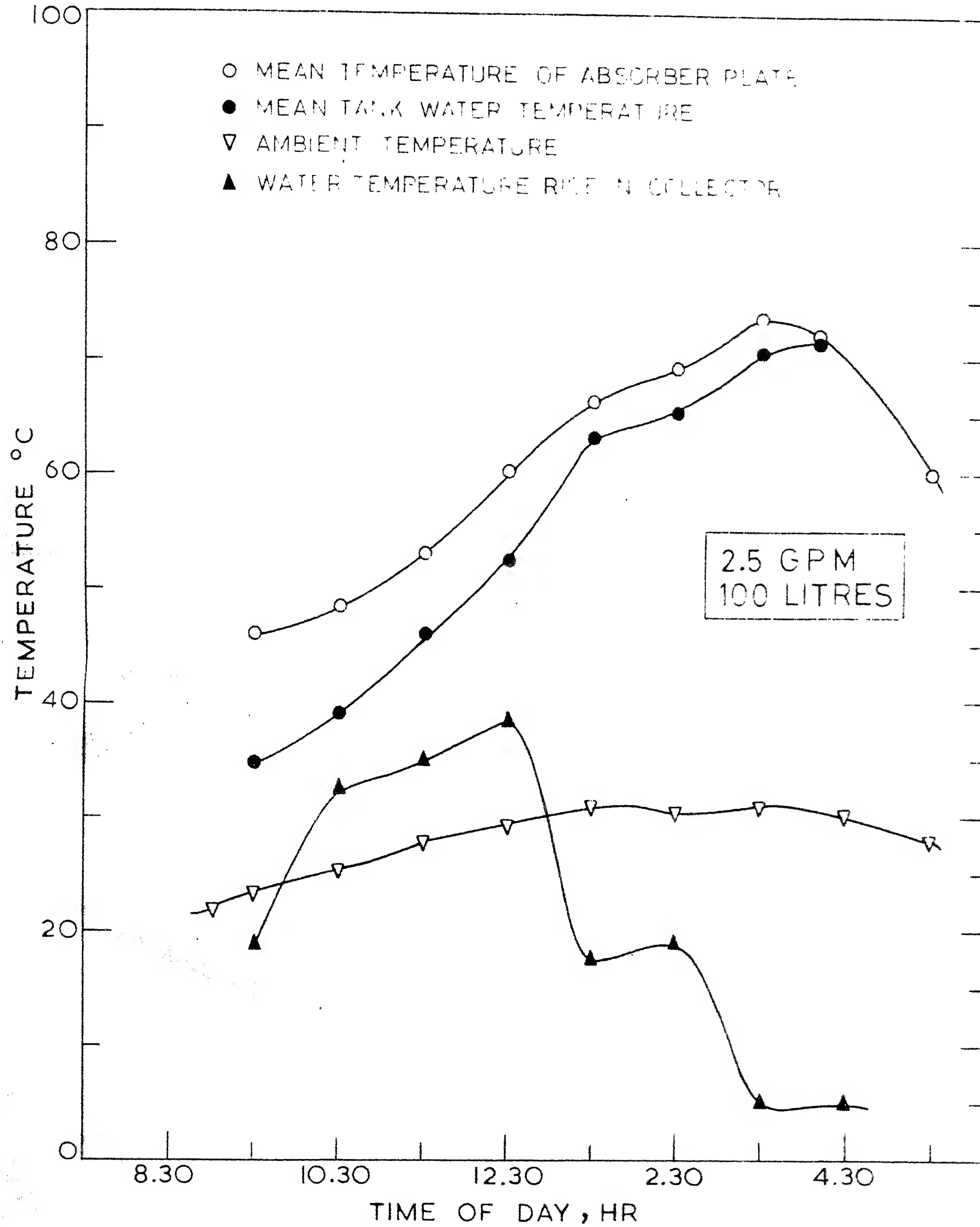


FIG. 4.5 EXPERIMENTAL RESULTS, MARCH 8,76

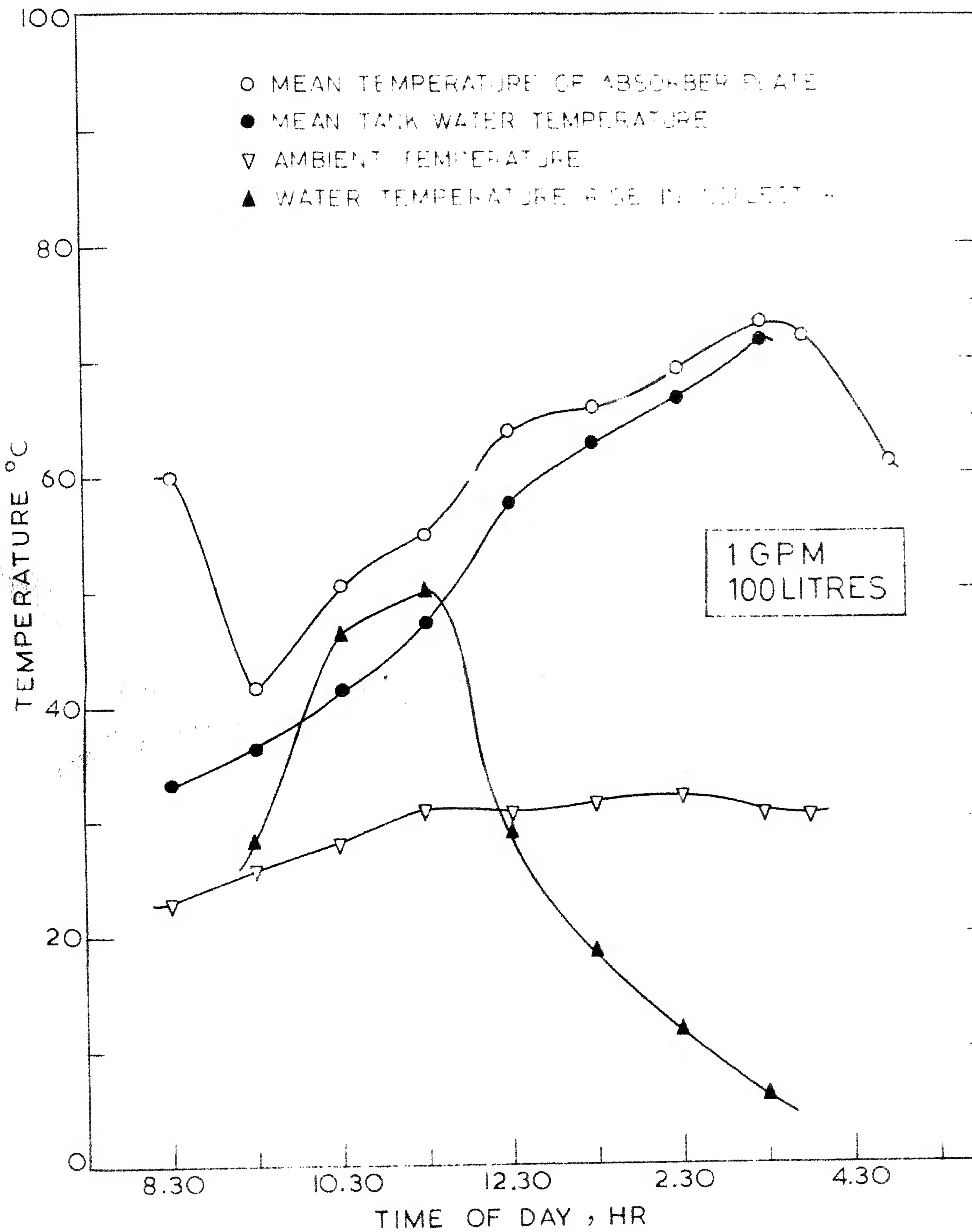


FIG. 4.6 EXPERIMENTAL RESULTS, MARCH 13, 76

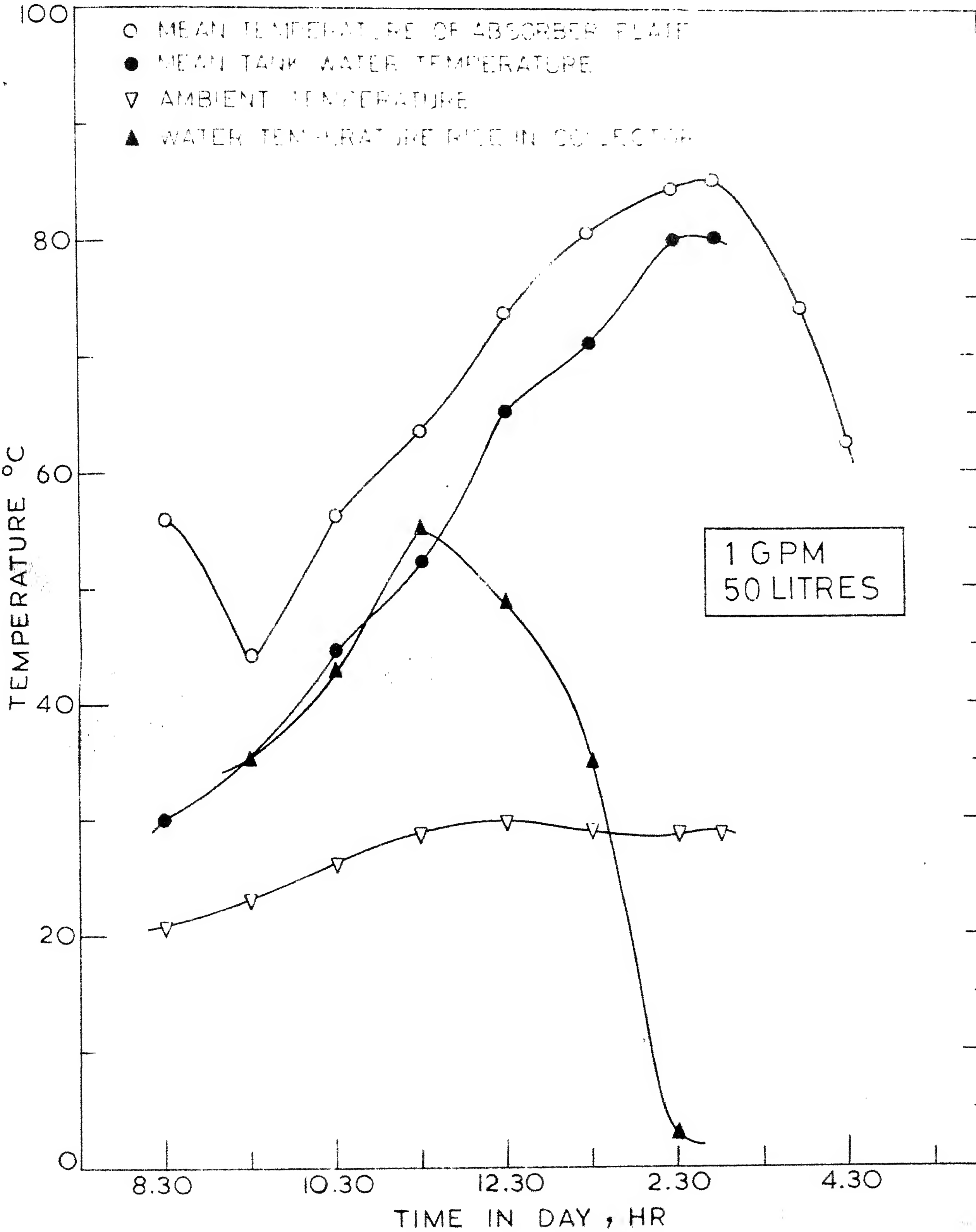


FIG. 4.7 EXPERIMENTAL RESULTS, MARCH 14,7

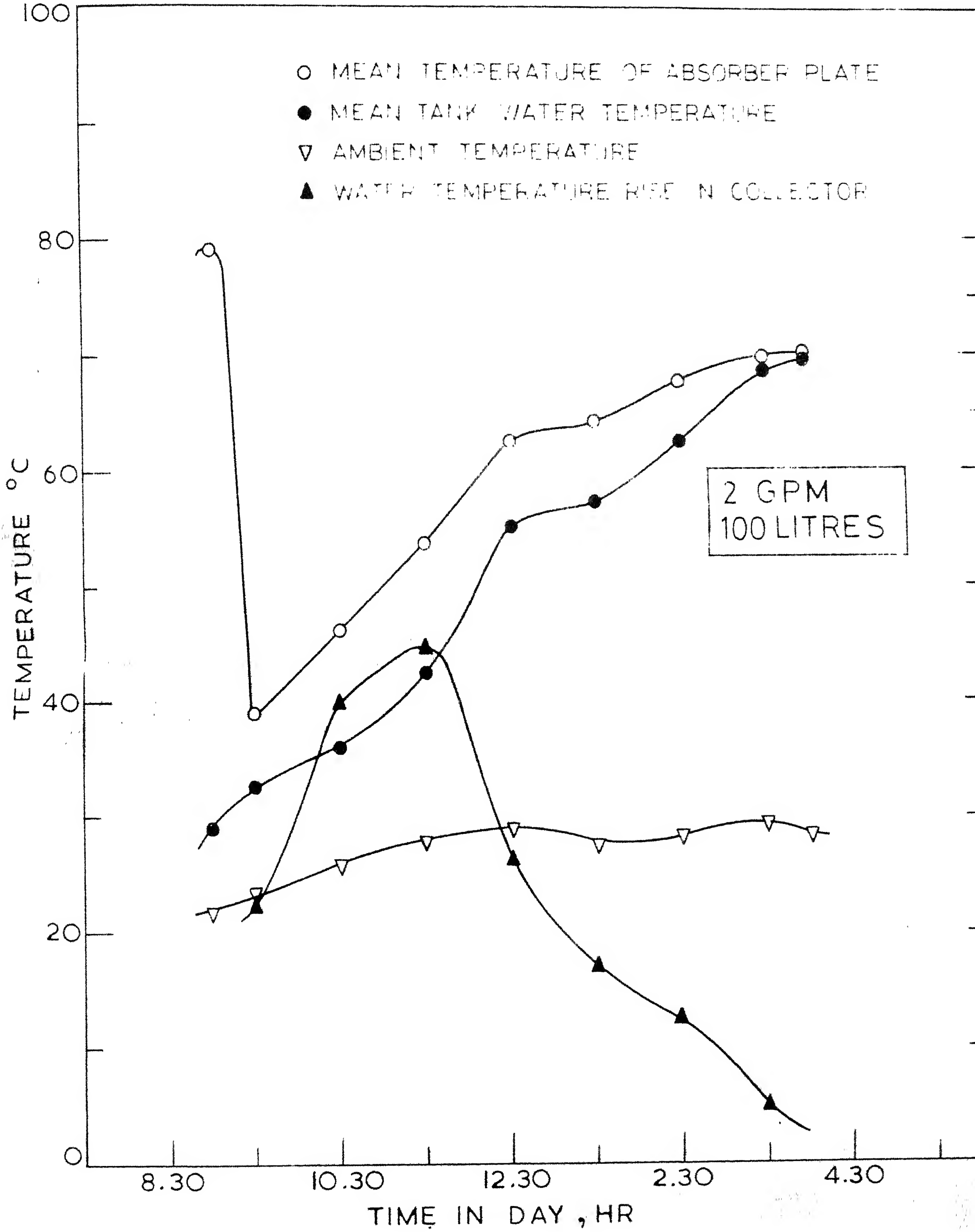


FIG. 4.8 EXPERIMENTAL RESULTS , MARCH 15 ,76

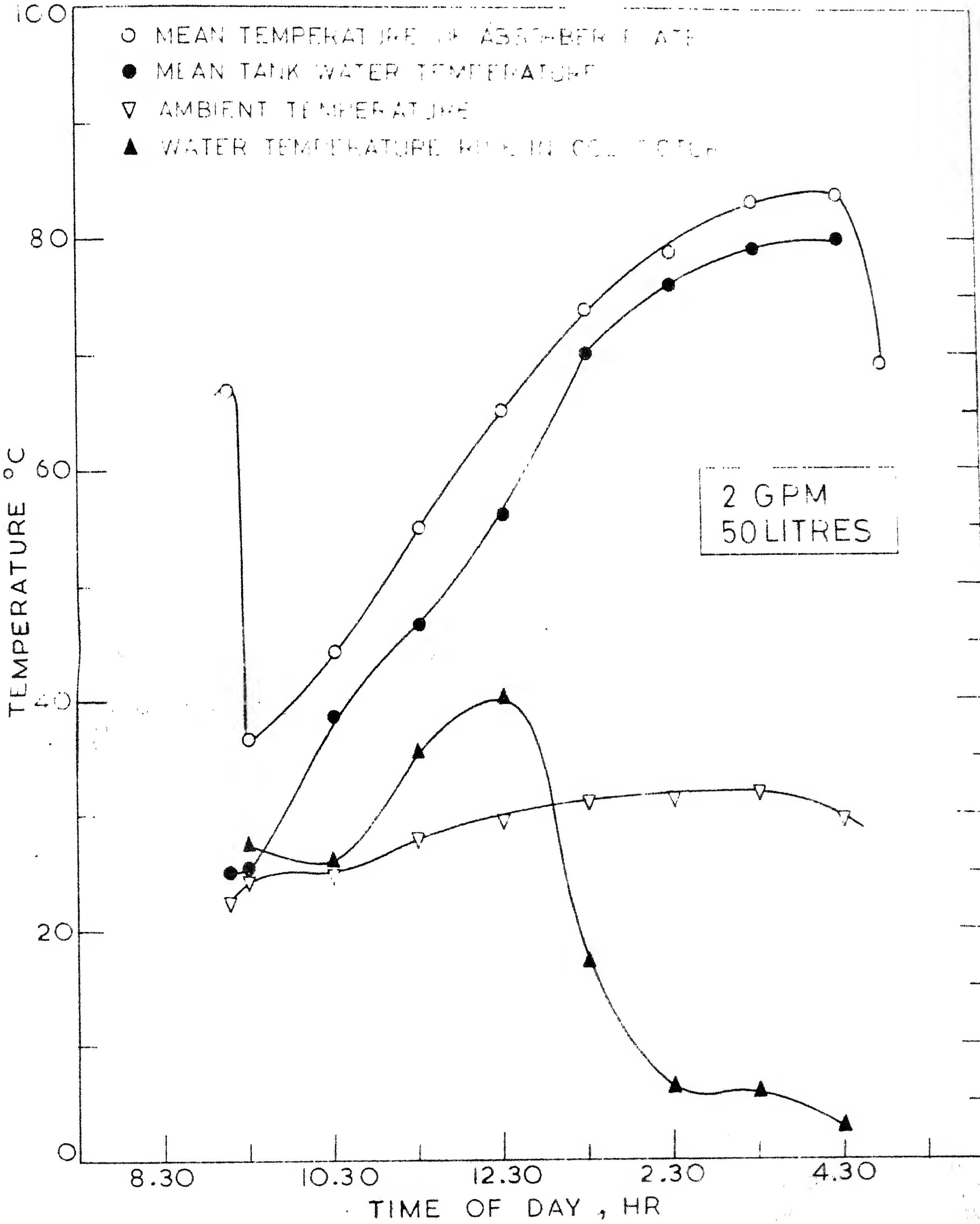


FIG. 4.9 EXPERIMENTAL RESULTS, MARCH 16, 76.

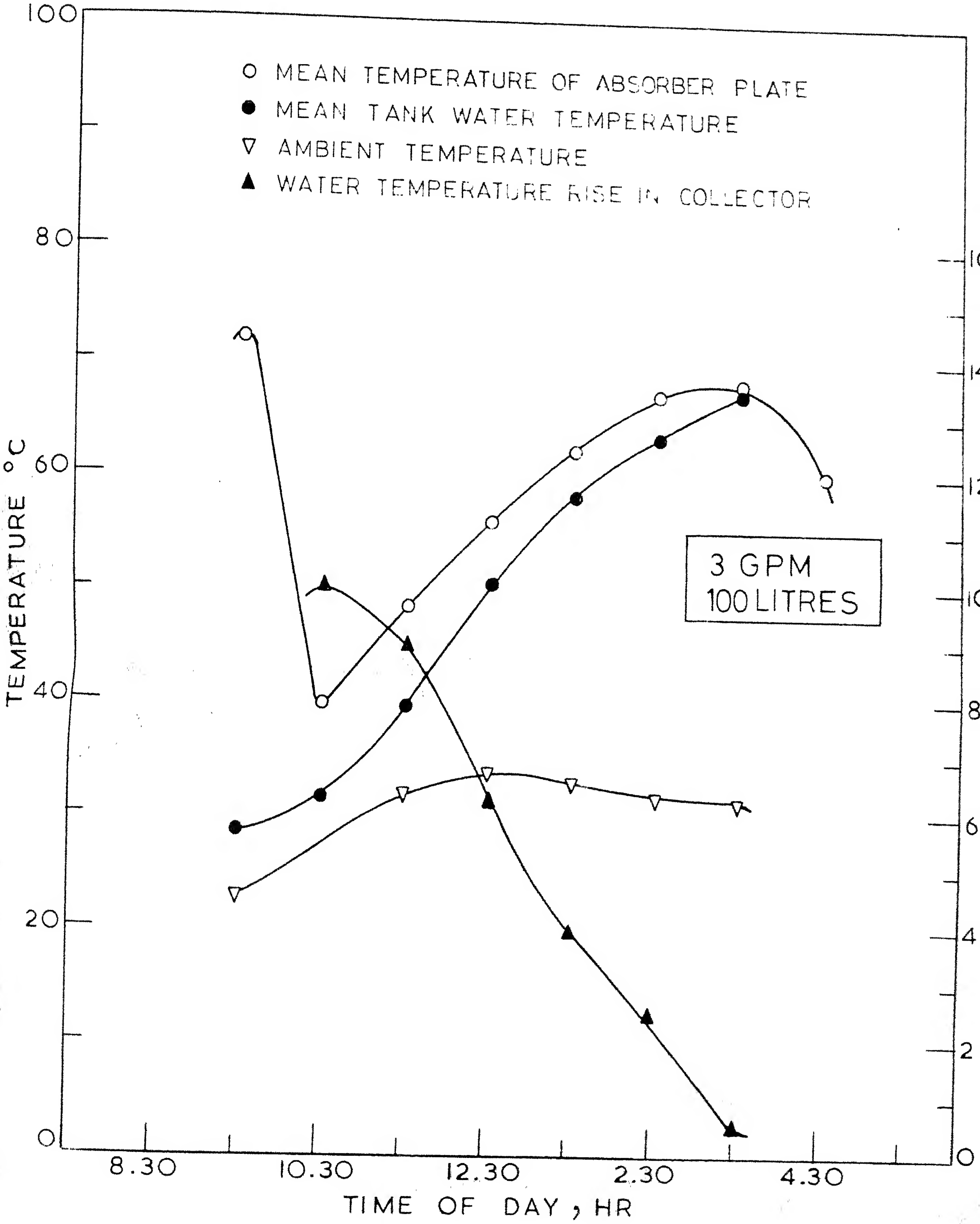


FIG. 4.10 EXPERIMENTAL RESULTS, MARCH 17, 76.

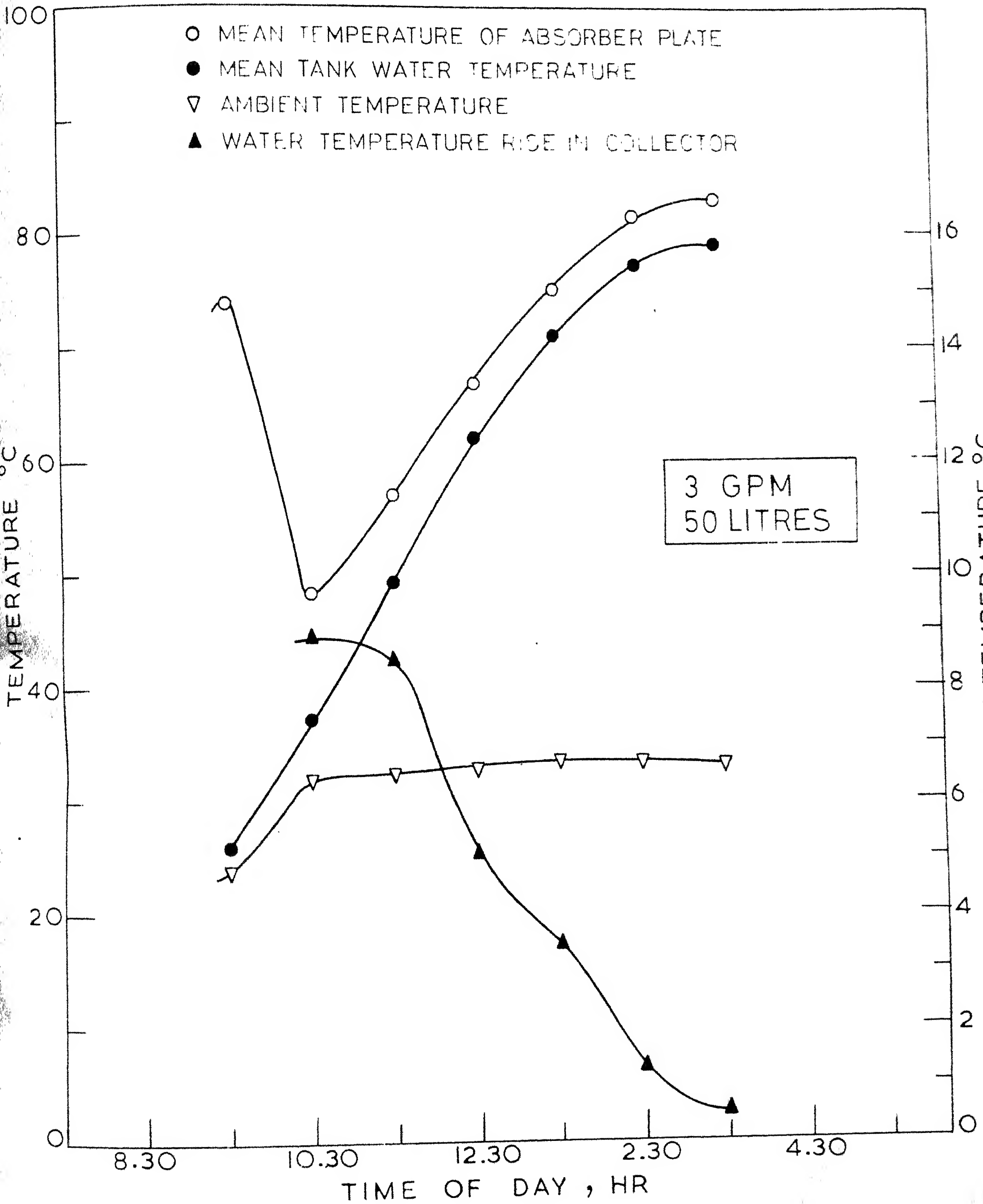


FIG. 4.11 EXPERIMENTAL RESULTS, MARCH 19, 76.

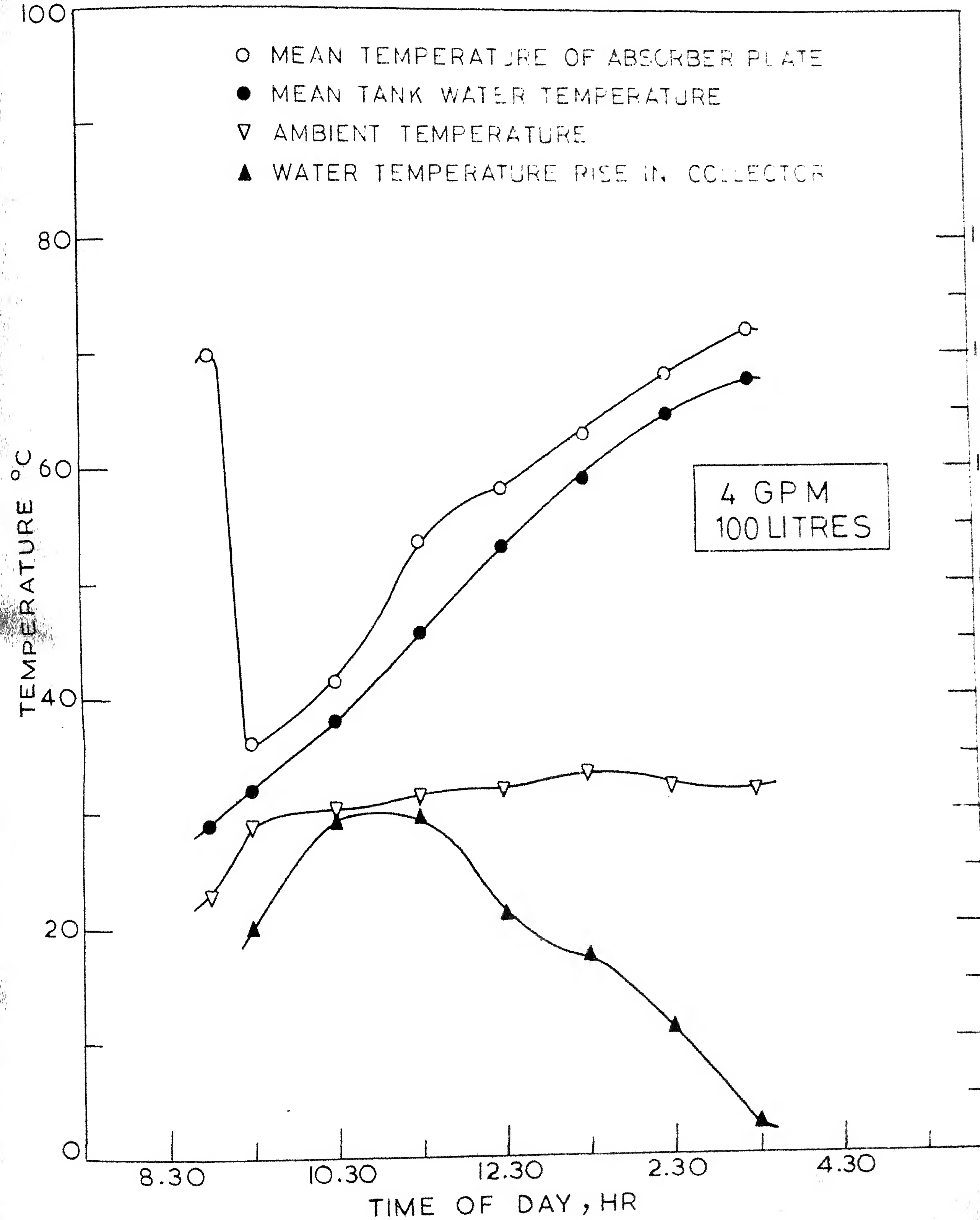


FIG. 4.12 EXPERIMENTAL RESULTS, MARCH 20, 76.

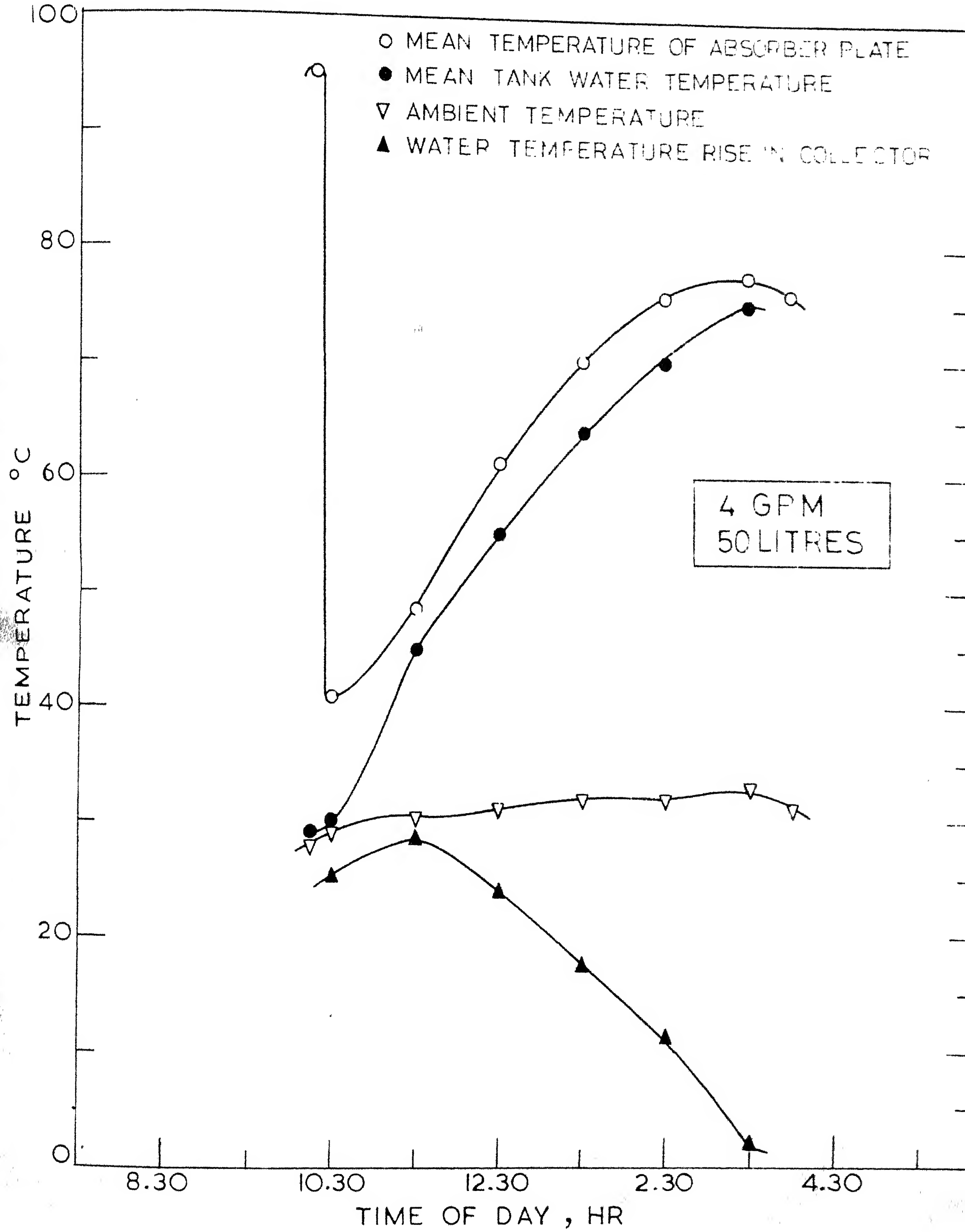


FIG. 4.13 EXPERIMENTAL RESULTS , MARCH 21, 76.

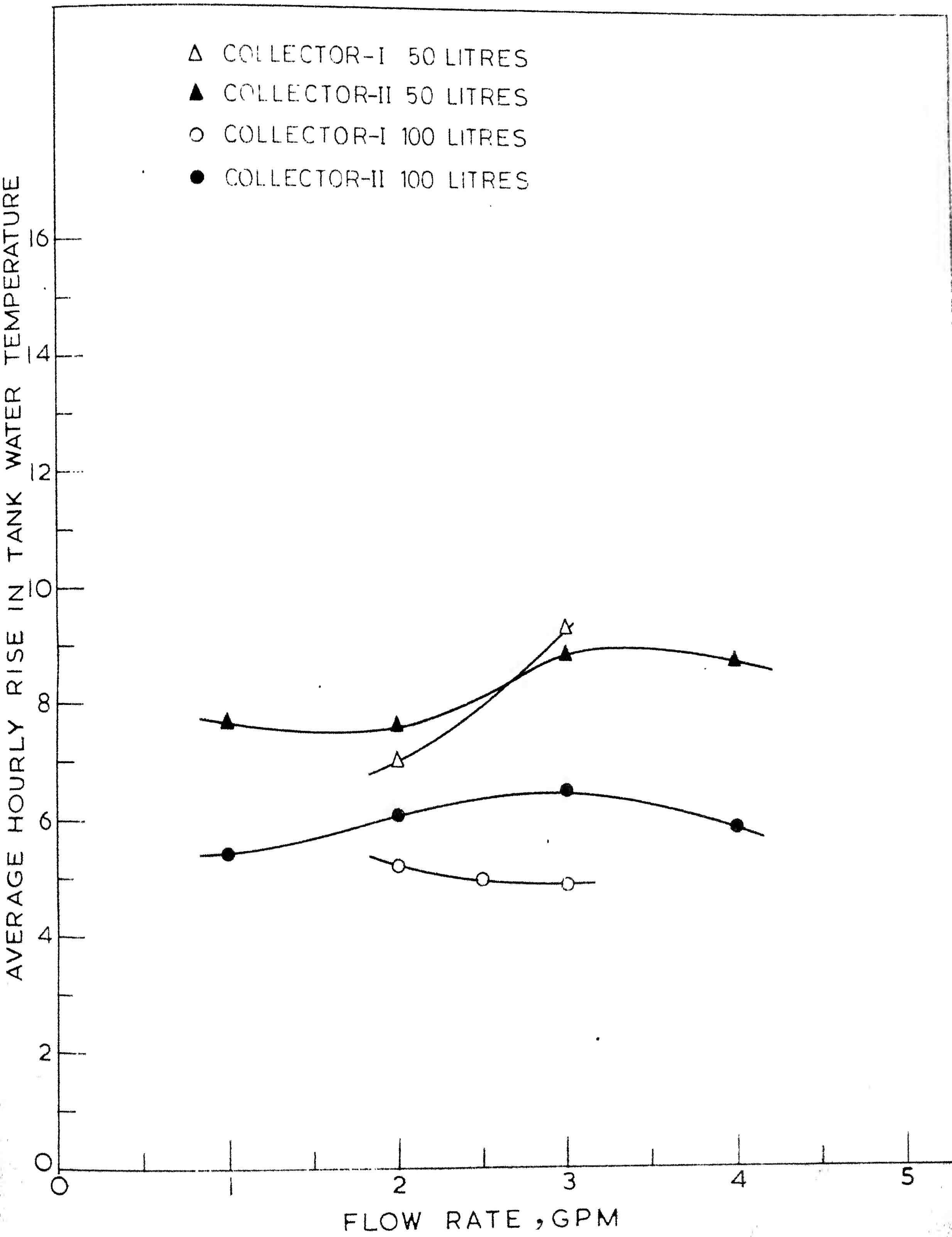


FIG. 4.14 EFFECT OF SYSTEM LOAD AND FLOW RATE ON AVERAGE RISE IN TANK WATER TEMPERATURE.

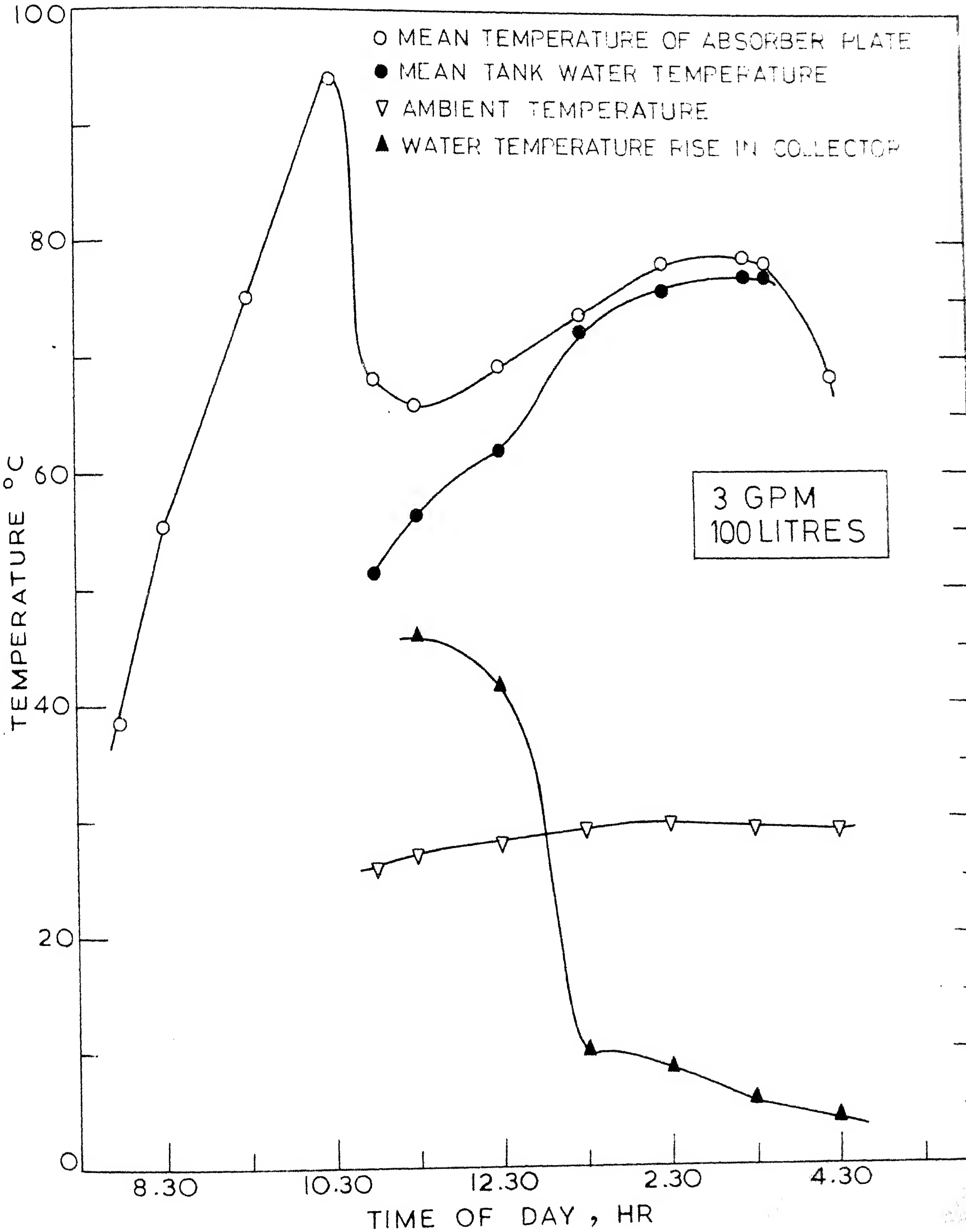


FIG. 4.15 EXPERIMENTAL RESULTS, MARCH 3,76.

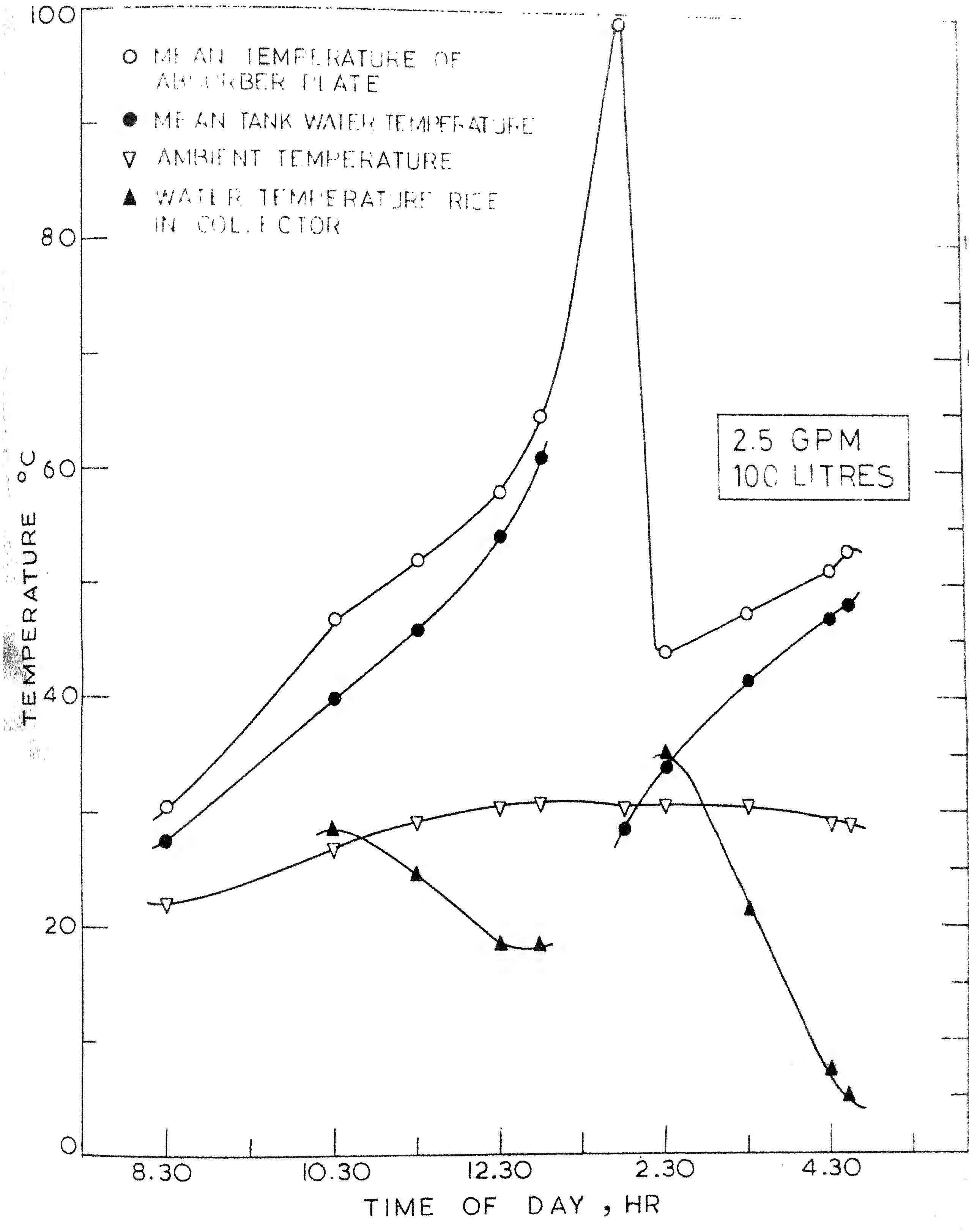


FIG. 4.16 EXPERIMENTAL RESULTS, MARCH 2, 76

28,5 °C and heated for 2.45 hours (2.0 p.m. to 4.45 p.m.). The temperature attained by this water is 48.0 °C and the heat gain from the absorber is 1950 kcal. The total amount of heat, thus, gained by 200 litres of water is 5300 kcal which would be much more as compared to the heat gain from the collector by a system load of either 100 or 200 litres of water.

This suggests that in order to utilise the collector more efficiently, stepwise heating of water as above or heating with intermittent draw-offs should be performed provided highest temperature attainment is not the goal.

Table 4.2 shows the storage characteristics of the insulated tank. Drop in temperature of the tank water varies considerably, from night to night as ambient temperature and initial hot water temperature which are two main parameters also vary from night to night. From the table it can be observed that temperature drop is more, when the temperature of hot water is relatively higher at the begining of the cooling cycle. At low initial temperature of the hot water, the drop is also very low. This suggests that the procedure of theoretical estimation of the insulation thickness that we have adopted is inadequate and needs improvement.

TABLE 4.2 Energy Storage Characteristics of the Tank

Date	Time	Ambient temp. Ta °C	Tank water temp. Tm °C	Storage period in hours	Drop in temp.
2.3.'76	4.30 p.m.	25.3	70.0		
3.3.'76	10.30 a.m.	24.5	53.0	18	17
3.3.'76	3.45 p.m.	29.0	77.5		
4.3.'76	6.30 a.m.	13.0	68.0	14.45	9
4.3.'76	3.30 p.m.	33.0	77.0		
5.3.'76	7.00 a.m.	11.0	64.0	15.30	13
5.3.'76	4.30 p.m.	31.8	68.4		
6.3.'76	7.00 a.m.	15.0	64.0	14.30	4.
6.3.'76	4.30 p.m.	31.0	79.0		
7.3.'76	7.25 a.m.	15.0	69.0	14.55	10.
7.3.'76	4.45 p.m.	29.0	48.0		
8.3.'76	7.30 a.m.	15.0	45.0	14.45	3.
15.3.'76	3.30 p.m.	29.5	68.5		
16.3.'76	6.30 a.m.	18.0	63.0	15	5.
16.3.'76	4.30 p.m.	29.8	80.0		
17.3.'76	6.40 a.m.	18.5	65.0	14.10	15.
17.3.'76	3.30 p.m.	31.2	67.5		
18.3.'76	6.30 a.m.	16.5	62.0	15	5.

4.3 COST ANALYSIS

The material used for the fabrication of the spray-type collector is all indigeneous and easily available in the local market. An approximate cost of each item for both collectors-I & II is given below.

Item	Cost in Rupees	
	Collector-I	Collector-
G.I. Sheets (3.25 m^2)	115.00	115.00
Nut, Bolts & Washers	20.00	20.00
Gasket and Shellac	15.00	15.00
Soldering	30.00	30.00
Nozzles	40.00	20.00
Pipes	30.00	7.00
Pipe fittings	20.00	10.00
Glass cover	30.00	30.00
Wooden casing	50.00	50.00
Plywood sheet	10.00	10.00
Insulation	50.00	50.00
Storage tank	85.00	85.00
Tank fittings	20.00	20.00
Total	515.00	462.00

The labour charges for fabrication cannot exceed Rs. 60.00 for two days period for two technicians to complete the unit. Thus the total cost of the unit will be around Rs. 575.00 for Collector-I and Rs. 525.00 for Collector II. This justifies preferring of Collector- over Collector-I both with the view point of economy and

4.4 CONCLUSIONS

The following conclusions may be drawn on the basis of the present study on spray type flat plate collectors wit water as the working fluid.

1. It is feasible to use spray-type flat plate collectors for forced circulation systems where large hot water demands are required.

2. In view of the economy, simplicity in design and performance, the spray-type collectors may be preferred over other existing designs.

3. Comparing between the two different nozzle configurations that have been studied in Collectors - I and II, no significant difference has been noticed in the maximum temperature attained by water. Collector - II however, is a simplified design containing less number of nozzles and piping and is, therefore, recommended over Collector - I.

4. The best performance has been attained for Collector - at 1 GPM flow rate for 50 litres of water as system load. The maximum temperature of water attained is 80 ^{o}C. This justifies our design criteria of using 1 m^2 of absorber area for heating 50 litres of water for a temperature rise of 50 ^{o}C. The study, however, was also extended for 100 litres system load with the

same absorber area. The maximum temperature of water, obtained is 71.6 °C at 1 GPM flow rate.

5. So far as the average hourly rise of tank water temperature is concerned, Collector - II gives the best performance at 3 GPM flow rate of water for both 100 litres and 50 litres of system load.

6. The same collector may be used for system loads higher than that for which it is designed, utilizing more heat from the absorber plate, though with less overall rise in fluid temperature.

4.5 SCOPE FOR FUTURE WORK

It is necessary to take up further work on spray-type collectors before a final marketable design is arrived at, for large scale water heating and other applications. The work needs study, mainly in the following directions.

1. To choose still a better nozzle arrangement retaining best performance and maximum efficiency and to further, reduce the overall cost of the collector.

2. A comparative study should be taken up with different collector designs simultaneously under the same operating and weather conditions.

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APPENDIX A

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$JOB    MEG286,TIME008,PAGES01 ,MAH   R.K.SHAH   M.TECH. (MECH.)  321515
$IBJOB          MAP
$IBFTC MAIN

C      COMPUTER PROGRAM FOR OPTIMUM TILT ANGLE

       COMMON HRS1, HRS2, MONTH1, MONTH2, NEELA, HASTU, DELPA
       REAL IGLOB(24,24), IDIFF(24,24), HANGLE(24), D(24), LATD
       INTEGER HRS1, HRS2, MONTH1, MONTH2, RITA(24), MONA(12), NAME(24),
      1DELPA(5,1  ), HOUR(24)
       DATA HOUR/6H 1 A.M,6H 2 A.M,6H 3 A.M,6H 4 A.M,6H 5 A.M,6H 6 A.M,6H
      2 7 A.M,6H 8 A.M,6H 9 A.M,6H10 A.M,6H11 A.M,6H12 A.M,6H1 P.MH,6H2 P
      3.M.,6H3 P.M.,6H4 P.M.,6H5 P.M., 6H6 P.M.,6H7 P.M.,6H8 P.M.,6H9 P.M
      4.,6H10P.M.,6H11P.M.,6H12P.M./
       DATA MONA/4HJAN.,4HFEB.,5HMARCH,5HAPRIL,3HMAY.4HJUNE,4HJULY,4HAUG.
      5,5HSEPT.,4HOCT.,4HNOV.,4HDEC./
       DO 11 KAMLA = 1,12
       NAM  (KAMLA) = MONA (KAMLA)
 11    NAME (KAMLA+12) = MONA (KAMLA)
       CALL HEMA
       READ 12, HRS1, HRS2, MONTH1, MONTH2
       IF (MONTH2.LT.MONTH1) MONTH2 = MONTH2+12
 12    FORMAT (5I2)
       READ 24, (HANGLE(I), I=HRS1, HRS2)
       READ 24, (D(I), I=MONTH1, MONTH2)
       DO 14 I = HRS1,HRS2
 14    HANGLE(I) = HANGLE(I)*3.1416/180.0
       DO 15 I = MONTH1,MONTH2
 15    D(I) = D(I)*3.1416/180.0
       DO 13 KK = 13,24
 13    D(KK) = D(KK-12)
       LEENA = 0
 16    LEENA = LEENA+1
       IF (LEENA.GT.12) STOP
       DO 17 I=1,12
```

```
      IGLOB(I,J) = 0.0
17    IDIFF(I,J) = 0.0
      READ 18, FITA
18    FORMAT (2 A4)
      PRINT 19, FITA
19    FORMAT (/2 X,2 A4)
      PRINT 20
20    FORMAT(//11X,*MONTHS*,16X,*TIME *,15 ,*B OPTIMUM*/)
      DO 21 I = MONTH1,MONTH2
      READ 24, (IGLOB(I,J),J=HRS1,HRS2)
21    CONTINUE
      DO 22 I = MONTH1,MONTH2
      READ 24, (IDIFF(I,J), J = HRS1,HRS2)
22    CONTINUE
      DO 23 KK = 13,24
      DO 23 JJ = 1,24
      IGLOB(KK,JJ) = IGLOB(KK-12,JJ)
23    IDIFF(KL,JJ) = IDIFF )KK-12.A+)
      READ 24, LATD, RO
      LATD = LATD*3.1416/180.0
24    FORMAT (16F5.2)
25    CALL MALINI
      IF (MONTH2.LT.MONTH1) MONTH2 = MONTH2+12
      IF (NEGLA.NE.0) GO TO 16
      X1 = 0.0
      Y2 = 0.00
      DO 26 I = MONTH1,MONTH2
      DO 26 J = HRS1,HRS2
      SINEB = COS(LATD)*COS(HANGLE(J))*COS(D(I))+SIN(LATD)*SIN(D(I))
      FORM = SIN(LATD)*COS(HANGLE(J))*COS(D(I))-COS(LATD)*SIN(D(I))
      X1=X1+(IGLOB(I,J)-IDIFF(I,J))*FORM/SINEB
26    Y2=Y2+(IGLOB(I,J)-IDIFF(I,J))+IDIFF(I,J)*0.379
      Y2Y2=X1/Y2
      BOPT=ATAN(Y2Y2)*180.0/3.1416
      PRINT 27, NAME (MONTH1),NAME (MONTH2), HOUR (HRS1), HOUR (HRS2),
     7OPT
27    FORMAT(/16X,A6,* TO *,A6,3X,A6,*. TO *,A6,F16.2)
      GO TO 25
```

```
$IBFTC HEMA
      SUBROUTINE HEMA
      COMMON HRS1, HRS2, MONTH1, MONTH2, [illegible]EL1, [illegible]CD, [illegible]G1
      INTEGER DEEPA(5,1[illegible])
      I = 1
 28   READ 29, (DEEPA(J,I), J=1,5)
 29   FORMAT(5I2)
      IF (DEEPA(5,I).NE.[illegible]) GO TO [illegible]
      I = I+1
      GO TO 28
 30   NEETU =
      RETURN
      END
$IBFTC MALINI
      SUBROUTINE MALINI
      COMMON HRS1, HRS2, MONTH1, MONTH2, NEELA. NEECD, DEEPA
      INTEGER DEEPA(5,1[illegible])
      NEETU = NEETU+1
      INTEGER HRS1,HRS2
      HRS1 = DEEPA(1,NEETU)
      HRS2 = DEEPA(2,NEETU)
      MONTH1 = DEEPA(3,NEETU)
      MONTH2 = DEEPA(4,NEETU)
      NEELA = DEEPA(5,NEETU)
      IF (NEELA.NE.2) NEETU=[illegible]
      RETURN
      END
$ENTRY
```

```
$JOB    MEG286,TIM= 3,PAG SU1 ,NAM=  R.K.SHAH    M.TECH. (MECH.)  3205
$IBJOB
$IBFTC MAIN

C      COMPUTER PROGRAM FOR THE EVALUATION OF SPECIFIC AREAS

       REAL ID(24,24), IDSMAL(24,24),HANGLE(24), D(24),LATD,B(24)
       INTEGER HRS1,HRS2,MONTH1,MONTH2,TEMP(20),RITA(20)
       DO540I=1,12
       DO540J=1,24
       ID(I,J)=0.0
540    IDSMAL(I,J)=0.0
       READ1,TEMP
   1   FORMAT(20A4)
       PRINT2,TEMP
   2   FORMAT(/20X,20A4)
       READ 22, RITA
  22   FORMAT(20A4)
       PRINT 25,RITA
  25   FORMAT(/20X,20A4)
       READ5,HRS1,HRS2,MONTH1,MONTH2
   5   FORMAT(5I2)
       DO4I=MONTH1,MONTH2
       READ5,(ID(I,J),J=HRS1,HRS2)
   4   CONTINUE
       DO6I=MONTH1,MONTH2
       READ5,(IDSMAL(I,J),J=HRS1,HRS2)
   6   CONTINU
       DO 555 KK=13,24
       DO555JJ=1,24
       ID(KK,JJ)=ID(KK-12,JJ)
 555   IDSMAL(KK,JJ)=IDSMAL(KK-12,JJ)
       READ5,(HANGLE(I),I=HRS1,HRS2)
```